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TECHNICAL REPORT NO. 10969

Investigation of the Factors Involved in Steering Tracklaying Vehicles

C-1225

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MAY, 1970

CONTRACT NO. DA-AE07-68-C-1225

J. R. RAY

by ALLISON DIVISION, GM

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U.S. ARMY TANK AUTOMOTIVE COMMAND, Warren, Michigan

ATAC TECHNICAL REPORT NO. 10969

INVESTIGATION OF THE FACTORS INVOLVED IN STEERING TRACKLAYING VEHICLES

FINAL REPORT

J. R. Ray

May 31, 1970

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Prepared by
Allison Division of General Motors
Indianapolis, Indiana

For
U S Army Tank Automotive Command
Warren, Michigan

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CONTRACT NO. DA-AE07-68-C-1225

INVESTIGATION OF THE FACTORS INVOLVED IN STEERING TRACKLAYING VEHICLES

Contract No. DA AE07-68-C-1225 and Modification Nos. 1, 2, and 3

Period Covered February 23, 1968, through January 31, 1970

I. INTRODUCTION

This report covers a current study of the forces involved and the power required in steering tracklaying vehicles. In the past steering system requirements have been reviewed periodically as the state-of-the-art has advanced and vehicle performance has improved. Because existing systems have been improved and new systems have been developed, it is necessary to reevaluate the factors which influence the steering forces which are translated into demands on the vehicle steer system in particular and the power package in general. In order that future vehicle performance may be predicted precisely, improvement and refinement of existing equipment may continue, and development of new systems may be undertaken, formulae have been reviewed and confirmed or revised as new test data have indicated.

A series of vehicle tests was conducted using the following vehicle-transmission combinations:

- Modified M109 vehicle (X500-1 transmission)
- Modified T48 vehicle (X700-1XT transmission)
- Modified M51 vehicle (XHM1500-1 transmission)

Although one vehicle adaptable to the three transmissions would have provided more compatible data, existing instrumentation in the three vehicles justified their usage. The data gathered were reduced electronically, and the results were analyzed for new findings.

In order to make the results of this program beneficial to those who will be developing tracked vehicle components and estimating vehicle performance, the report has been organized into the following sections:

- Theory of Steering a Tracklaying Vehicle
- Tracked Vehicle Steering Calculations
- Test Program - Investigation of Steering Forces

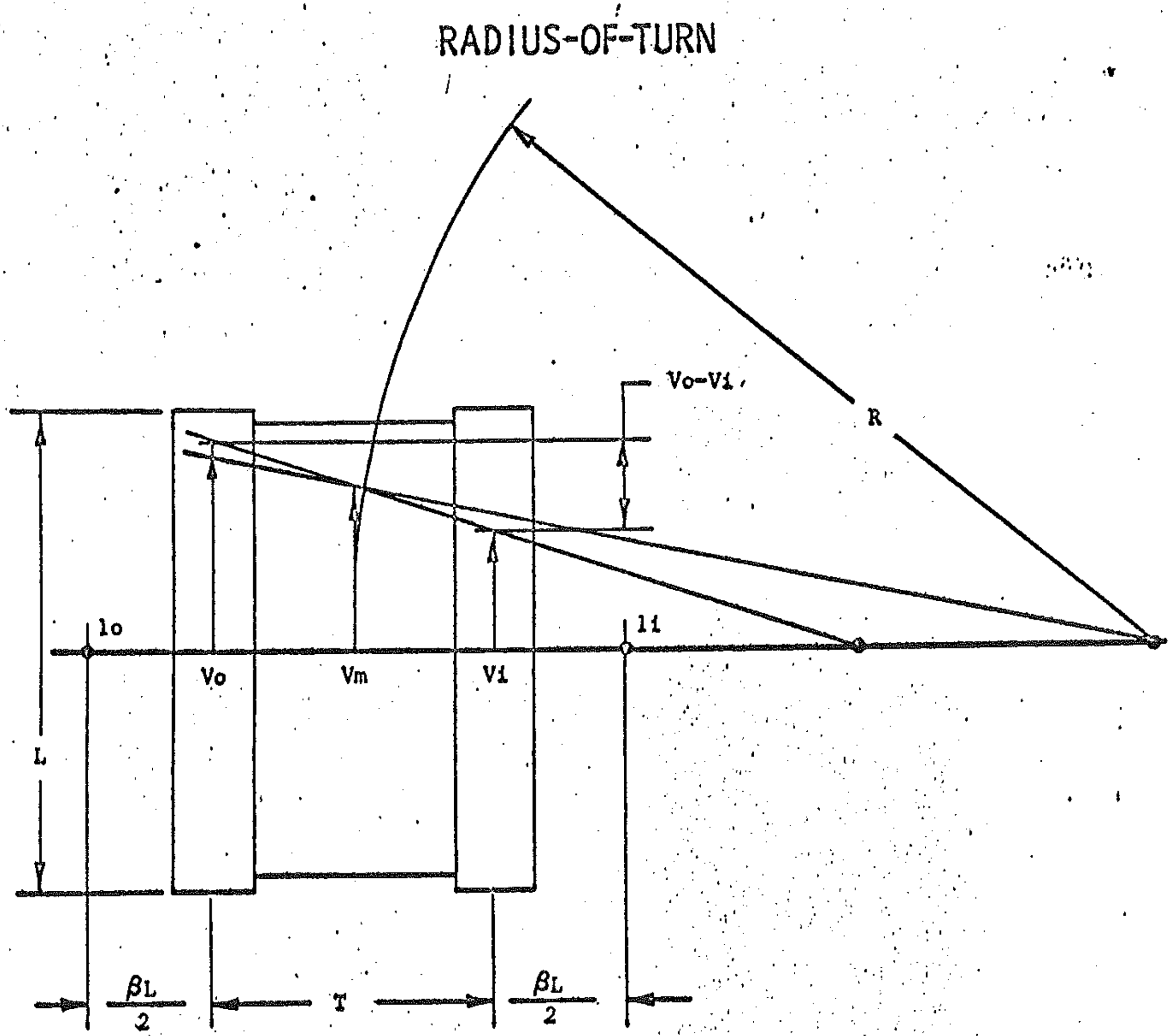

Fig. 1

II. THEORY OF STEERING A TRACKLAYING VEHICLE

A. The Tracklaying Path

The path of a tracklaying vehicle is a function of track speed difference and the geometry of the tracks themselves, that is, ground-contact length and spread of tracks. The classic or simplified analysis of the forces involved in steering a tracklaying vehicle[1] assumes that the tracks are essentially rigid bars, uniformly loaded and in uniform frictional contact with an impervious surface. On this basis, it is possible to describe mathematically the path traversed by a vehicle and the forces involved when the tracks are rotated at different speeds with respect to each other.

However, in actual practice, the tracks are not completely rigid, but are composed of individual track blocks which flex and pivot with respect to one another thereby permitting the track to lay a curved path. Furthermore, as explained by the Elasticity Principle[2], elastic deformation occurs between the suspension system, track bushings, blocks, and ground over which the track passes. This permits some lateral movement of the vehicle with respect to the ground before skidding of the tracks occurs. Thus, the track forces actually required, particularly when turns of large radii are made, are reduced when compared to those calculated. The closer a vehicle can follow a "free-turn" curved path, that is a path where no lateral skidding of the tracks occurs, the lower the required steer force will be (ref: Section IIIB).

B. Track Forces

The basic analytical method for determining the force required to steer a tracklaying vehicle is the Merritt formula:

$$F = K\mu W$$

where:

F is the longitudinal force applied at the track just sufficient to overcome the adhesion resistance to turning

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[1] The Evolution of a Tank Transmission, H. E. Merritt, The Institution of Mechanical Engineers, Volume 154 p 412, April 1946

[2] Investigation of Track Laying Vehicle Steer Requirements, M. D. Smith, Final Report Contract No. DA-33-008-ORD 1545 Detroit Arsenal, June 1958

K is a proportionality factor which takes into account the geometry of the vehicle (ref: Fig. 6, page 16 for a plot of K versus α, the reciprocal of the L/T ratio)

μ is the ground coefficient of adhesion

W is the weight of the vehicle supported by one track

But according to Merrit's formula, the turning effort required at any speed is a constant value regardless of the turning radius. If this characteristic were true, it would be very difficult to steer a high speed vehicle since, once enough bias torque to turn was obtained, the vehicle might turn on any radius thus presenting an unstable steer control.

However, experience has shown that the actual force required for large radii turns is considerably less than the force required for tighter turns. This characteristic of track force being inversely proportional to the radius of turn differs from Merritt's expression for estimating turning effort requirements. This is the more desirable position, for as long as there is increasing torque for a sharper turn, the vehicle will be more controllable.

Even so, under certain topographical conditions the required track force for maintaining a steer may decrease. In such an instance, an over-steer can develop. Should the steer turning couple substantially exceed the vehicle traction stability, the vehicle will continue turning and its polar velocity will continue to accelerate.

Also, when a driver initiates a turn, he wants quick results and tends to apply a high turning couple. This couple overcomes track resistance and vehicle polar inertia, thus the vehicle starts accelerating its polar velocity and turn rate. As the track blocks start sliding, their traction coefficient drops making the vehicle easier to turn. If the vehicle is turning with some forward velocity, centrifugal force reduces the required turning force due to the longitudinal difference between the vehicle centers of gravity and friction. Therefore, as the vehicle reaches the desired turn rate, the driver must reduce the steer force or he will continue to turn more sharply.

In summation, the maximum forces encountered are those required to enforce the minimum turning radius at the maximum traction condition, and the forces required to initiate a turn are always greater than those required to sustain the turn. The important point for both of these conditions is that a force-controlled system in which track force difference is employed to enforce turning, is inherently unstable and, to a degree, uncontrollable.

C. Power

From a practical standpoint, steering a tracklaying vehicle is rotating the vehicle about its center of gravity while translating it forward.

Rotating the vehicle about its center is analagous to slipping a clutch, in that the tracks skid over the ground. The power to accomplish this action, then, is a function of vehicle weight, vehicle geometry, ground coefficient of adhesion, and rotational speed of the vehicle about its polar axis. Note however, that this power - which is consumed at the tracks - is not a function, and is independent, of the method employed to produce the differential track speed.

The steer system, when enforcing a differential track speed - that is reducing the inner track speed below that of the outer track speed - absorbs a significant amount of power which is generated by dragging the inner track over the ground. If the vehicle is to maintain momentum and speed, it must compensate for this power loss by providing additional power to the outer track. This makeup power can come either from:

- transfer of the inner track, absorbed power,
- additional engine power, or
- a combination of the above.

Feedback power, then, is a function of track forces, vehicle speed, track speed potential difference, and the efficiency of the transmission feedback path.

These two power requirements and the power required for normal propulsion make up the total requirement for conducting a steer maneuver.

With transmission systems having a means of connecting the inner and outer tracks (mechanically, electrically, or hydraulically) feedback power can be transmitted to the outer-track - in varying degrees. In those systems where the tracks are mechanically geared together and variations in ratio are produced by slipping a grounding device, a percentage of feedback power will be absorbed at the slipping surface when less than maximum ratio is selected. In such instances less feedback power will be available to the outer track and makeup power requirements will be high. Although the total power consumed can remain constant, the percentages absorbed at the

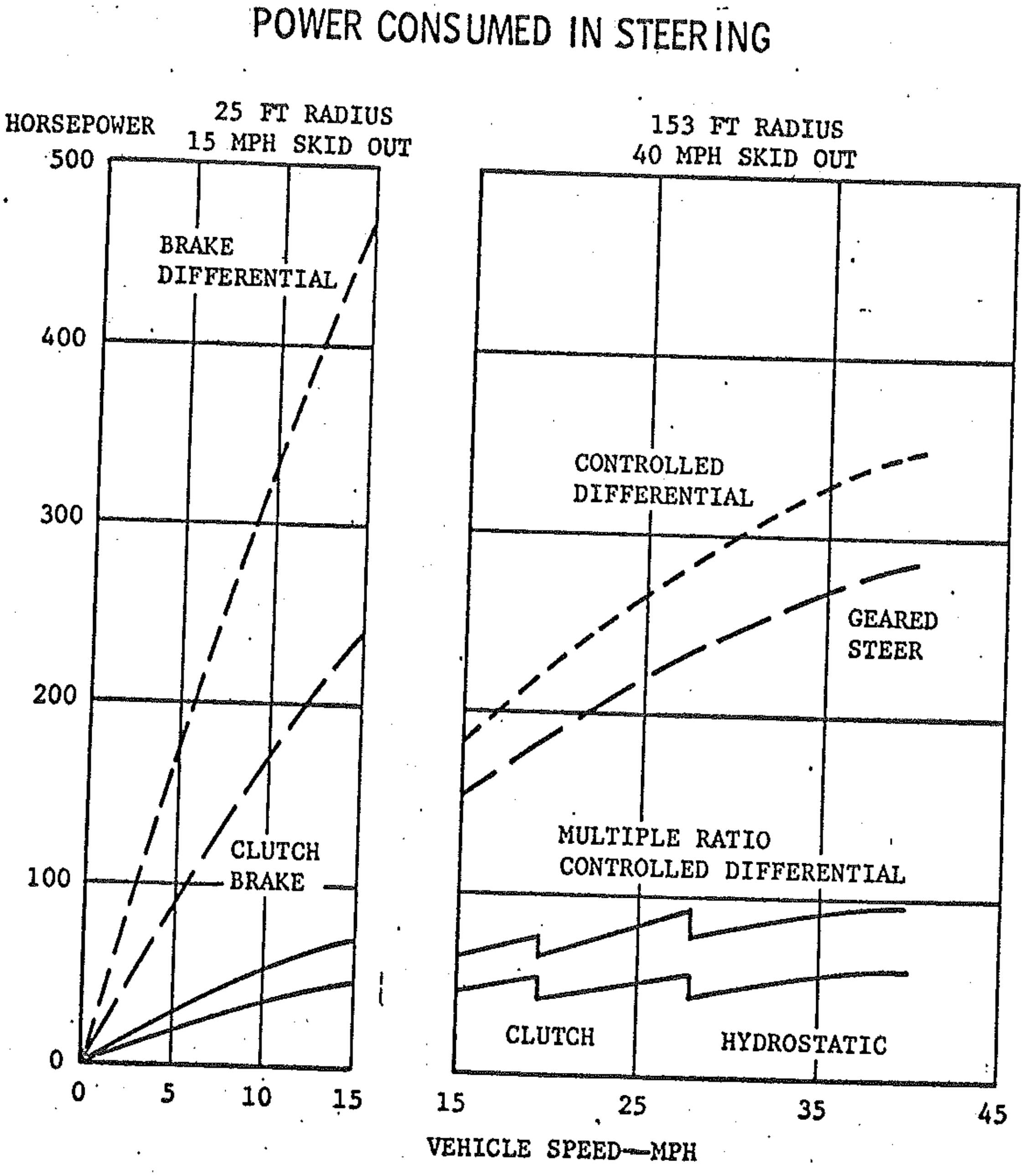

POWER CONSUMED IN STEERING

Fig. 2

slipping device and the outer track may vary. An indicator of the amount of power expended in any slipping clutch steering device may be found in the difference between the actual speed difference and the potential speed difference of the tracks.

It follows then that a steering system which offers multiple steer ratios, inversely proportional to transmission gear ranges, will be more efficient during steering than a single ratio system (as shown in Fig. 2 opposite). For example:

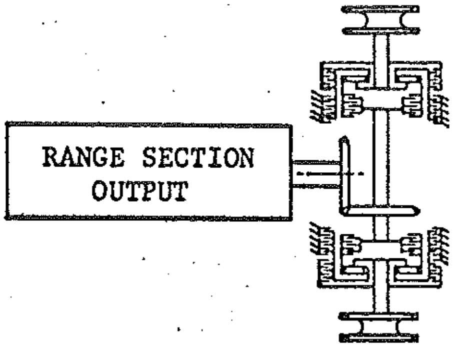
- The geared steer system is fully regenerative; however, it does not have optimum steer. It, in effect, downshifts the inner track at high speeds to make use of vehicle momentum, but it does not increase the speed of the outer track. A clutch-brake device may be used to back up the road speed steer ratio for low speed maneuverability. But here all transferred power is wasted (engine power is not), making high speed usage extremely inefficient.

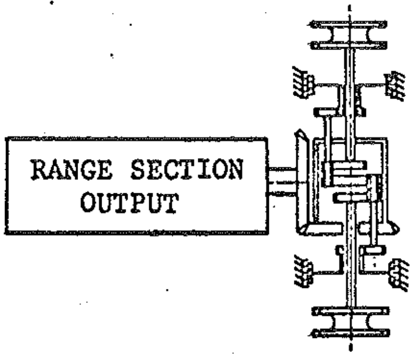
- The controlled differential system is also fully regenerative when the inner-track clutch is fully engaged. This is further enhanced by the fact that the speed of the outer track is increased in proportion to the decrease of the inner track speed. Thus vehicle speed is maintained. However, a differential system which offers only one steer ratio must compromise its high speed efficiency in order to turn tightly enough to satisfy low speed maneuverability. The differential can be braked to ground for extremely tight turns, but this is quite inefficient as half of the engine power plus all the transfer power is wasted while slipping.

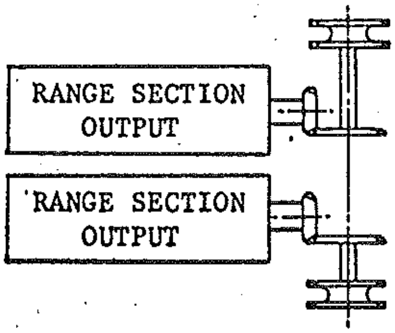
A clutch controlled differential system with multiple steer ratios combines the method of feedback of the single ratio controlled differential system with the step feature of multiple gear ranges. Thus the steering potential decreases with increasing vehicle speed (the minimum radius of turn is increased as the transmission is operated in successively higher propulsion gear ranges) for greater efficiency and improved controllability.

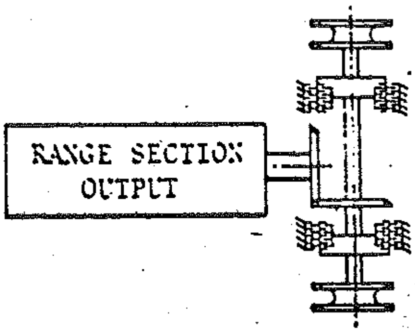
A hydrostatically controlled differential system with multiple steer ratios, due to the infinitely variable characteristics of the hydrostatic mechanism, has still better overall efficiency than the clutch-controlled system. It has an infinite number of maximum or "full" ratio positions which means that is is always at maximum efficiency, approximately 80 percent. The slipping

MODERN STEER SYSTEMS


GEARED STEER


CONTROLLED DIFFERENTIAL


INDEPENDENT PROPULSION


CLUTCH BRAKE

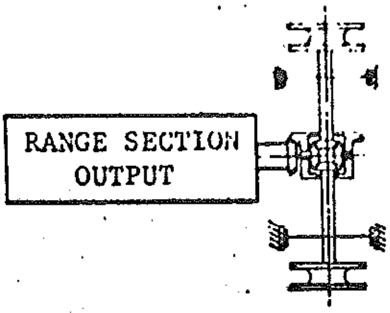

BRAKED DIFFERENTIAL

Fig. 3

clutch can achieve higher efficiency, 100 percent, when fully engaged. But it is normally less efficient, when slipping, than the hydrostatic mechanism. The infinitely variable hydrostatically controlled system also provides greater steer safety because, again, it more closely matches the steer potential in each range to the skid-out characteristics of tracked vehicles (shown in Fig. 4 overleaf).

- With independent propulsion systems such as the one shown opposite there is no feedback power from inner to outer track. However, in other systems where a single power source is split to two separate transmissions - such as the XHM-1500 used for the investigative portion of this program - regeneration can be achieved. In the case of the XHM-1500 the feedback is subject to the inefficiencies of two hydrostatic systems. Though the principle feature of this system is its infinitely variable speed ratio potential, the outer track speed does not necessarily increase in proportion to the decrease in inner track speed.

- The clutch-brake system is non-regenerative in that when slowdown of the inner track is required, all feedback power is absorbed by the brake. The main feature of this system is its relative simplicity.

- In the brake differential system the inner track brake absorbs all feedback plus up to half of the propulsive power. This is the simplest but least efficient of all steer systems.

D. L/T Ratio

There is no theoretical limit to the ratio of track contact length to tread width (L/T). So long as the tracks are separated by a physical distance on either side of the vehicle center of gravity, they have a mechanical advantage in performing a lateral skid by producing axial tractive forces. However, from the vehicle design standpoint the L/T ratio has considerable effect upon the effort required to steer.

For the short and wide vehicle (L/T ratios of 1:1 and under), the turning effort applied to the track has a long moment arm, whereas the track blocks sliding on the ground have a short moment arm. This vehicle is easy to turn, but it is very unstable. It will tend to turn off to a new tangent every time it hits a bump due to the "unwinding" of the loaded drive train, track, and suspension components.

Conversely, for the long, narrow vehicle (L/T ratios of 1.8:1

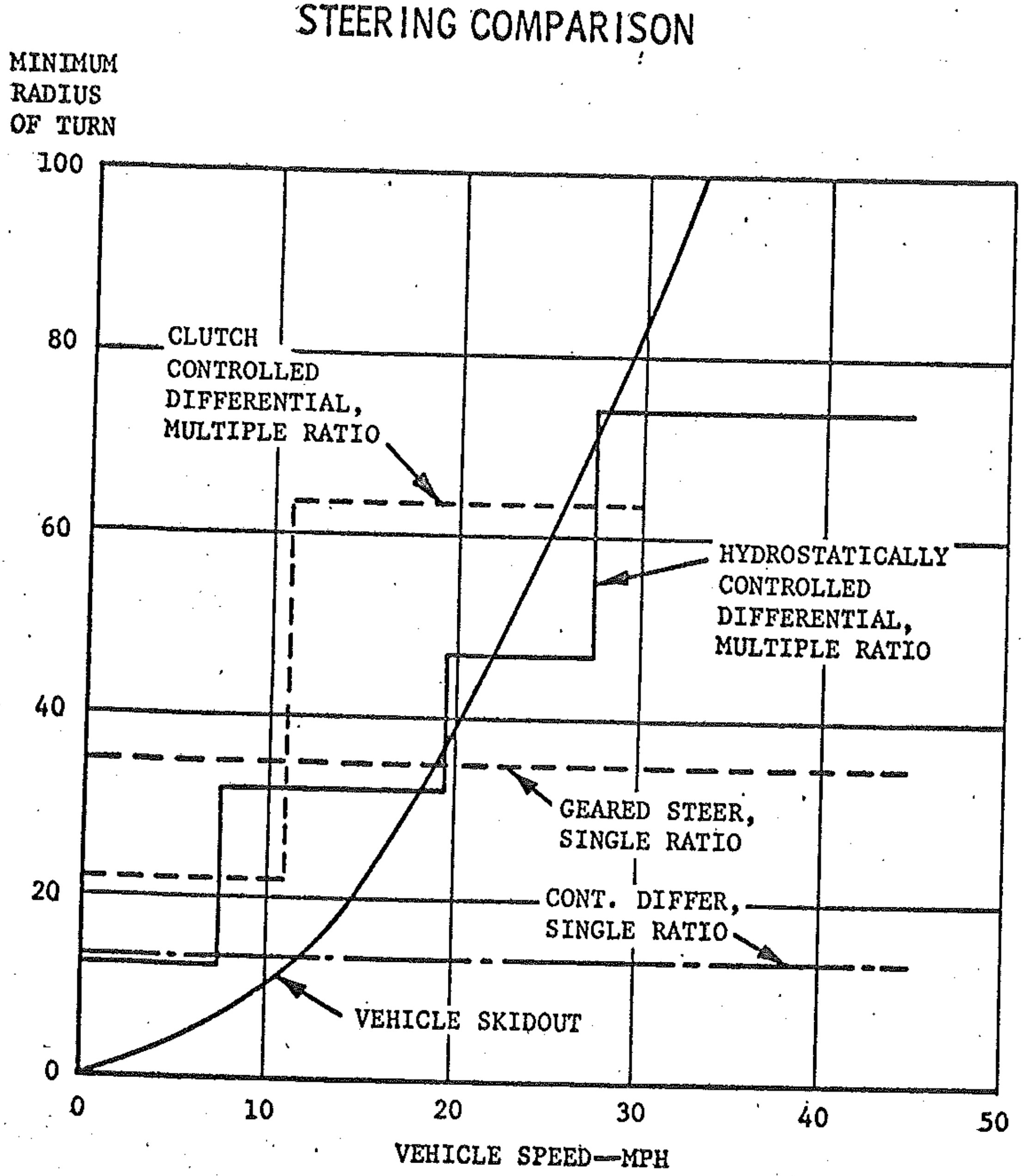

STEERING COMPARISON
MINIMUM RADIUS OF TURN
100
80
60
40
20
0
0
10
20
30
40
50
VEHICLE SPEED—MPH
CLUTCH CONTROLLED DIFFERENTIAL, MULTIPLE RATIO
HYDROSTATICALLY CONTROLLED DIFFERENTIAL, MULTIPLE RATIO
GEARED STEER, SINGLE RATIO
CONT. DIFFER, SINGLE RATIO
VEHICLE SKIDOUT

Fig. 4

and greater), the turning effort has a short moment arm and the track blocks at the ends have a long moment arm. This makes the vehicle difficult to turn since the steer force approaches the axial skid force and increases the minimum radius of turn in each range. In addition, there is a severe reduction of vehicle speed because the steering power is high (ref: Section IIIC). If a larger power plant is employed to offset these conditions, changes must also be made in the transmission to provide greater steer ratio and to accept the additional power. The end result is greater complexity in the transmission and an increase in the weight of the overall power package.

The foregoing steer difficulties with large L/T vehicles, while always a problem to some degree for operation on a hard surface, become acute problems under soft, muddy ground conditions. Here the vehicle tracks penetrate, and the resistance to slewing is the shear strength of the soil. Under such conditions vehicles with large L/T ratios may not be able to develop sufficient axial traction to overcome the resistance to turning, thus they quite often will not steer at all.

As previously noted, there are no theoretical limits to the L/T ratio. However, the more successful tracked vehicles have been designed within practical limits of 1.125:1 and 1.70:1.

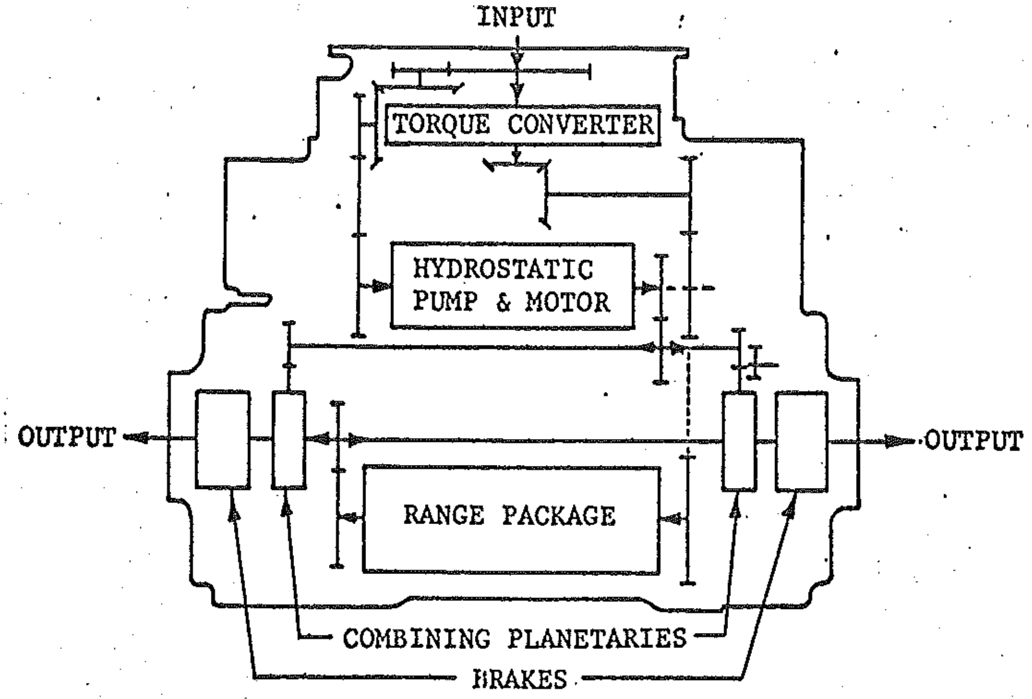
Note: With the advent of "speed control", wherein differential track speeds are enforced by a variable speed device, a general improvement in steer stability can be expected. This means, in turn, that steer quality equivalent to present levels can be maintained for vehicles with smaller L/T ratios. The end result, more importantly, will be greater steering efficiencies (reductions in steer power requirements).

E. Steer Control

Independent track drive systems, such as electric drives, hydrostatic drives, and multiple power packages, provide the optimum concept in terms of track speed variation for steering. However, from the standpoint of controls for both steer and propulsion functions, complexity of design, and efficiency of feedback power, the independent track drive is less practical than a differential steer system with hydrostatic control. The feedback power, for instance, of an independent track drive must pass through two complete systems and incur losses proportional to the complexity of the two. This includes: a mechanical, electrical, or hydraulic transfer system, the two mechanical output gear trains, and possibly both range and steer systems.

HYDROSTATIC STEER SYSTEM COMPARISON

ENGINE-DRIVEN SYSTEM



TURBINE-DRIVEN SYSTEM

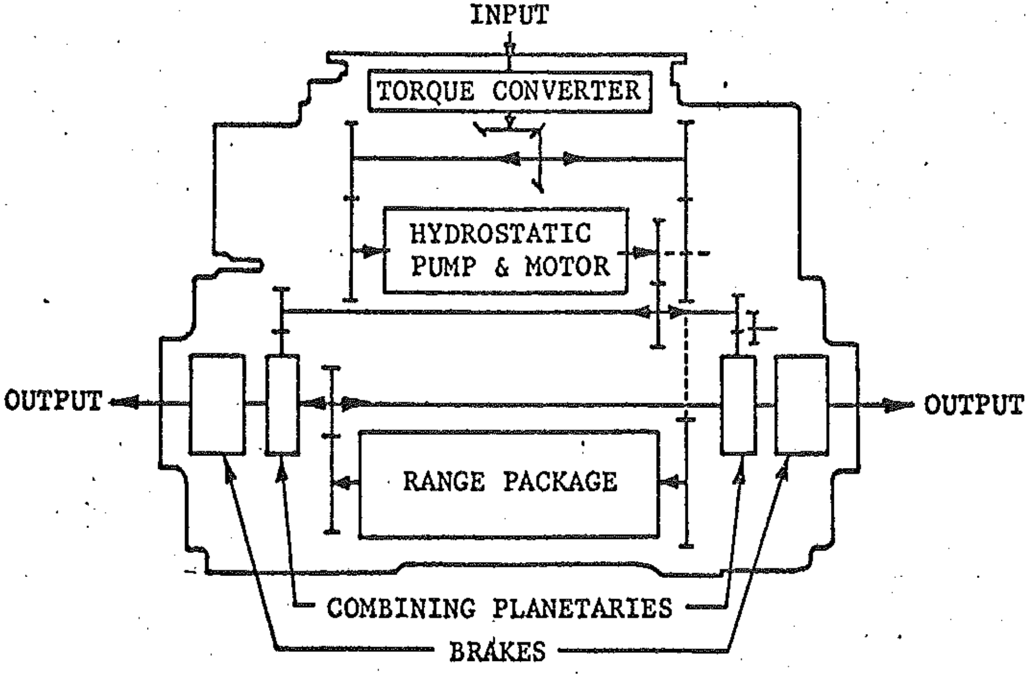


Fig. 5

With a hydrostatically controlled differential steer system, however, the transferred power path is from the carrier of the combining differential of the inner, or slower, track to the ring gear; then, from the ring gear of the combining differential of the outer, or faster, track to its output carrier. This affords a direct path which subjects the majority of the transferred power to only the mechanical loss of the combining planetary differentials. Only the reaction power takes the less efficient path through the steer differential train and the hydrostatic system.

Differences in quality of steer are also found within hydrostatically controlled differential steer systems. In engine-driven versus turbine-driven hydrostatic systems, for example (as shown in Fig. 5, opposite), when the transmission torque converter is in operation and:

- when the vehicle is under engine power, and engine-driven steer control system provides "spiral steer" capabilities whereas the turbine-driven steer system is limited to a fixed steer-ratio in each range. "Spiral steer" is available with an engine-driven steer system because when the unit is in converter drive, a high engine speed (hydrostatic pump speed) can be maintained. Thus, in turn, a high steer bias is also maintained. As the vehicle slows down, it is capable of tighter and tighter turns with eventual transition into pivot steer as the turbine comes to stall.

- when the vehicle is coasting (engine idling), engine-driven steer is found to be objectionable because of engine stalling during attempted steers at light throttle. This is caused by the vehicle steer torque requirements being greater than the torque produced by the engine at the light throttle position. Engine stalling usually occurs during close-quarter maneuvering. To overcome engine stalling, the driver must advance the throttle and apply the brakes to obtain a high engine speed before calling for a steer. This is objectionable in close quarters because the vehicle could accelerate rapidly upon release of the brakes if in gear.

  When coasting or being towed, with a dead engine, the engine-driven steer configuration suffers a complete loss of steer. Conversely, under the same conditions, the turbine-driven steer system provides full steer control down to a near stop.

When the torque converter is locked up, hydrostatic steer is the same for either engine- or turbine-driven steer control.

VEHICLE PROPORTIONALITY FACTORS

(REF: AN ANALYSIS OF THE FORCES INVOLVED IN STEERING A FULL TRACK LAYING VEHICLE, F. BLAIR JR., GENERAL MOTORS INSTITUTE, APRIL 1949)

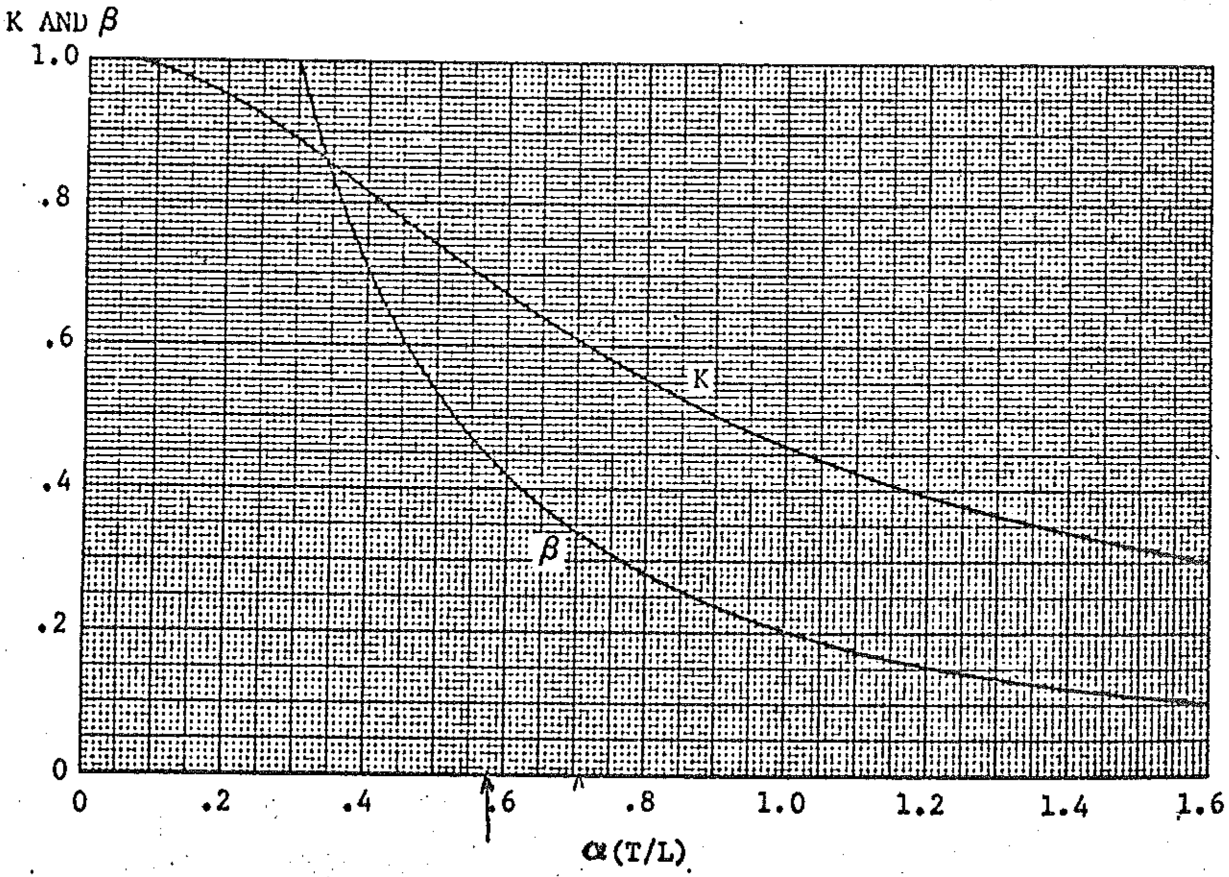


Fig. 6

III. TRACKED VEHICLE STEERING CALCULATIONS

The following subsections, A thru D, show example calculations for radius-of-turn, vehicle speed, steer forces, power requirements, and L/T ratios. The vehicle data given for those examples are those of the M109 test vehicle used as part of the steer study test program discussed in Section IV.

A. Radius-of-Turn and Vehicle Speed

For known or given values of inner and outer track speed, the radius of turn can be determined from the modified Merritt equation:

$$R = \left(\frac{V_o + V_i}{V_o - V_i} \right) \left(\frac{T + \beta L}{2} \right)$$

where:

T = Track center-to-center distance
L = Track length on ground
β = Vehicle proportionality factor (ref: Fig. 6, opp.)
V_o = Outer track velocity
V_i = Inner track velocity

Example

Given:

T = 109 in.
L = 154.5 in.
β = .336
V_o = 21.45 mph
V_i = 18.80 mph

$$R = \frac{109 + .336 \times 154.5}{2} \left(\frac{21.45 + 18.80}{21.45 - 18.80} \right)$$

$$= 1223 \text{ in. or } 102 \text{ ft}$$

Conversely, from the assumption that the forward vehicle speed is the arithmetic mean of the inner and outer track speeds, the following derived equations enable calculating track speeds when radius of turn and vehicle speed are provided.

$$V_o = V_m \left(1 + \frac{T + \beta L}{2R} \right)$$

$$V_i = V_m \left(1 - \frac{T + \beta L}{2R} \right)$$

RADIUS-OF-TURN VARIABLE

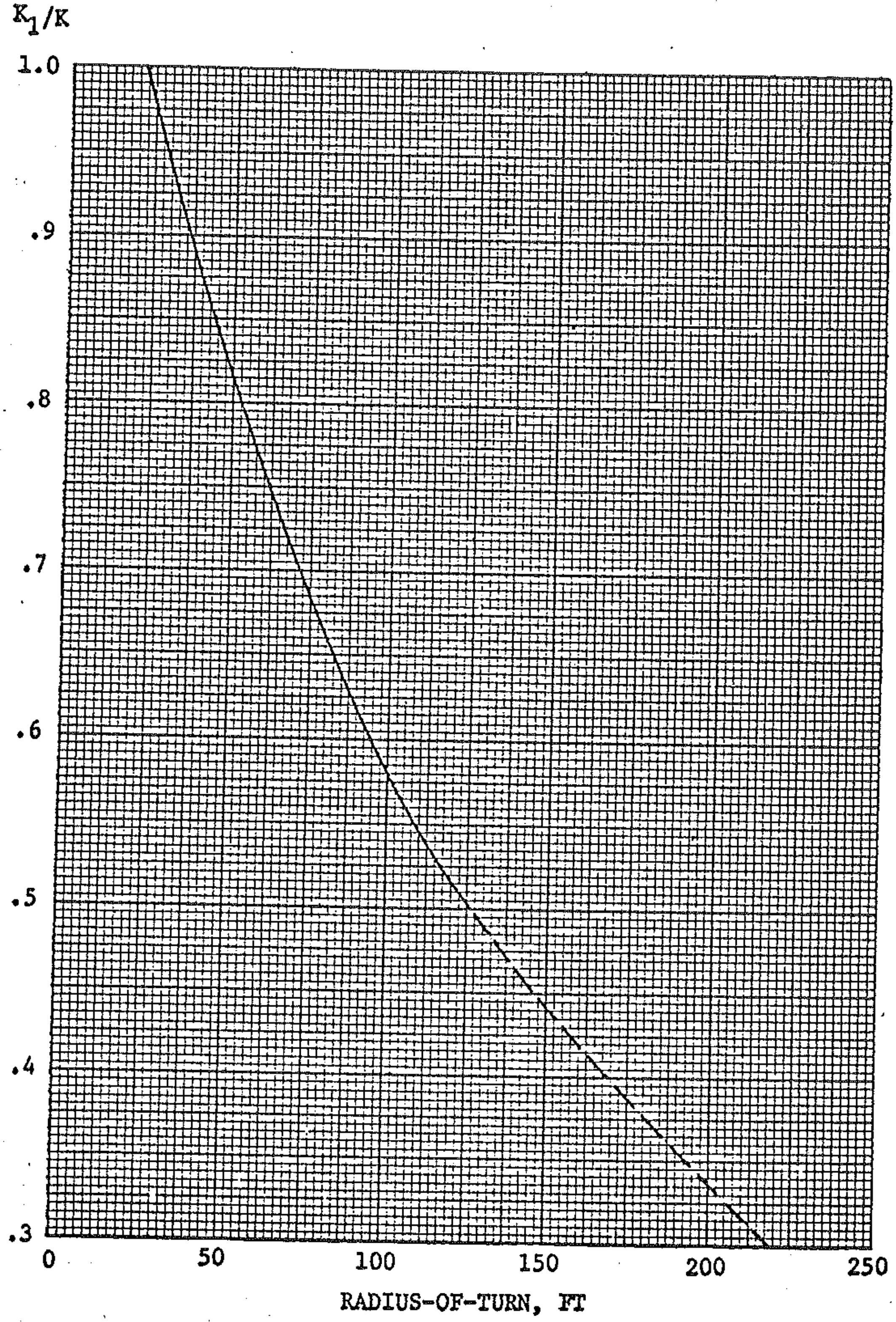


Fig. 7

where:

V_m = Mean vehicle speed

Example

Given:

$V_m = 20$ mph
$T = 109$ in.
$L = 154.5$ in.
$\beta = .336$
$R = 50$ ft

$$V_o = 20 \left(1 + \frac{109 + .336 \times 154.5}{2 \times 50 \times 12} \right)$$

$$= 20 (1 + .134)$$

$$= 22.68 \text{ mph}$$

$$V_i = 20 \left(1 - \frac{109 + .336 \times 154.5}{2 \times 50 \times 12} \right)$$

$$= 20 (1 - .134)$$

$$= 17.32 \text{ mph}$$

B. Forces During a Turn

As discussed in Section IIB, use of the basic Merritt equation for force, $F = K \mu W$, results in a constant value regardless of the radius of turn. This equation can be altered, however, to show the different forces required at different radii as the tracks attempt to flex or curve to match the vehicle's path. The factor K, a constant based simply on vehicle geometry, can be replaced with a variable, K_1. This variable takes the factor K and modifies it, on the basis of test data (ref: Section IV), to reflect the changes in force attendent to changes in radius of turn.

$$F = K_1 \mu W$$

where:

K_1 = Vehicle proportionality factor modified for radius of turn (ref: Fig. 7, opp.)
μ = Ground coefficient of adhesion
W = Gross vehicle weight

GLOSSARY OF TERMS

F_p = Resistance to propulsion, lb

F_s = Slewing force, lb

V_m = Mean vehicle speed, mph

ΔV = Track differential speed, mph

R = Radius of turn, ft

R_w = Wind resistance factor

R_r = Rolling resistance factor

R_g = Grade resistance factor

A = Vehicle frontal area, ft^2

W = Gross vehicle weight, lb

T = Track center-to-center distance, in.

L = Track length on ground, in.

β = Function of vehicle geometry (ref: Fig. 6, page 16)

K_1 = Vehicle proportionality factor, modified for radius of turn (ref: Fig. 7, page 18)

μ = Ground coefficient of adhesion

$Eff_{t\&fd}$ = Efficiency of transmission and final drives (through feedback path)

Example

Given:

$T = 109$ in. $R = 100$ ft, 50 ft
$L = 154.5$ in. $K_1 = .354,\ .495$
$W = 52{,}180$ lb
$\mu = .75$ $\downarrow$
$K = .608$ .608x58=

$F = .354 \times .75 \times 52{,}180$

$= 13{,}850$ lb @ 100 ft radius

$F = .495 \times .75 \times 52{,}180$

$= 19{,}300$ lb @ 50 ft radius

C. Power Requirements

The power required at the sprockets, HP_s, to drive a track-laying vehicle through a steer maneuver is the sum of several individual requirements - the power required for propulsion, HP_p, the power required for lateral track scrubbing, HP_{sc}, and the power required to compensate for inner track loss of momentum, $HP_{it} - HP_{fb}$. At the sprockets, this becomes:

$$HP_s = HP_p + HP_{sc} + (HP_{it} - HP_{fb})$$

$$\boxed{HP_p = \frac{F_p V}{375}}$$

$$\boxed{F_p = R_w A V^2 + (R_r + R_g)\, W/2000}$$

$$HP_{sc} = \frac{F_s \Delta V}{375}$$

$$F_s = K_1 \mu W/2$$

$$\Delta V = \frac{(T + \beta L)\, V}{12R}$$

$$HP_{it} = \boxed{\frac{F_s V}{375}}$$

$$HP_{fb} = HP_{it}\ Eff_{t\&fd}/100$$

(See the glossary, opposite, for definition of terms.)

Example

Given:

$T = 109$ in.	$\beta = .336$
$L = 154.5$ in	$K_1 = .495$ ($K = .608$)
$A = 84$ ft^2	$R_w = .0025$
$W = 52{,}180$ lb	$R_r = 85$ lb/ton
$V_m = 20$ mph	$R_g = 0$ lb/ton
$R = 50$ ft	$\mu = .75$
	$\mathrm{Eff}_{t\&fd} = 81\%$

Propulsive Power

$$F_p = R_w A V_m^2 + (R_r + R_g)\, W/2000$$

$$= .0025 \times 84 \times 20^2 + (85 + 0)\ 52{,}180/2000$$

$$= 2304 \text{ lb}$$

$$HP_p = \frac{F_p V_m}{375}$$

$$= \frac{2304 \times 20}{375}$$

$$= 123 \text{ horsepower}$$

Scrubbing Power

$$F_s = K_1 \mu W/2$$

$$= .495 \times .75 \times 52{,}180/2$$

$$= 9680 \text{ lb}$$

$$\Delta V = \frac{(T + \beta L)\ V_m}{12R}$$

$$= \frac{(109 + .336 \times 154.5)\ 20}{12 \times 50}$$

$$= 5.36 \text{ mph}$$

$$HP_{sc} = \frac{F_s \Delta V}{375}$$

$$= \frac{9680 \times 5.36}{375}$$

$$= 138 \text{ horsepower}$$

Momentum Loss Power

$$HP_{it} = \frac{F_s V_m}{375}$$

$$= \frac{9680 \times 20}{375}$$

$$= 517 \text{ horsepower}$$

$$HP_{fb} = HP_{it}\ Eff_{t\&fd}/100$$

$$= 517 \times 81/100$$

$$= 418 \text{ horsepower}$$

Total Power Requirement

$$HP_s = HP_p + HP_{sc} + (HP_{it} - HP_{fb})$$

$$= 123 + 138 + (517 - 418)$$

$$= 360 \text{ horsepower}$$

D. L/T Ratio

Where small L/T ratios are being considered, it is necessary to evaluate the stability of the vehicle during maximum steer maneuvers. Large L/T ratios, on the other hand, because they provide greater vehicle stability, require more steer effort thus more power.

Small L/T Ratios

If the point of maximum steer ratio attainable in each range is used, the maximum track speed differential is achieved and the minimum radius of turn can be calculated. In the equation:

$$R = \left(\frac{V_o + V_i}{V_o - V_i} \right) \left(\frac{T + \beta L}{2} \right)$$

If V_i is set at unity, then the factor $\left(\frac{V_o + V_i}{V_o - V_i} \right)$ can be

EFFECT OF VEHICLE L/T

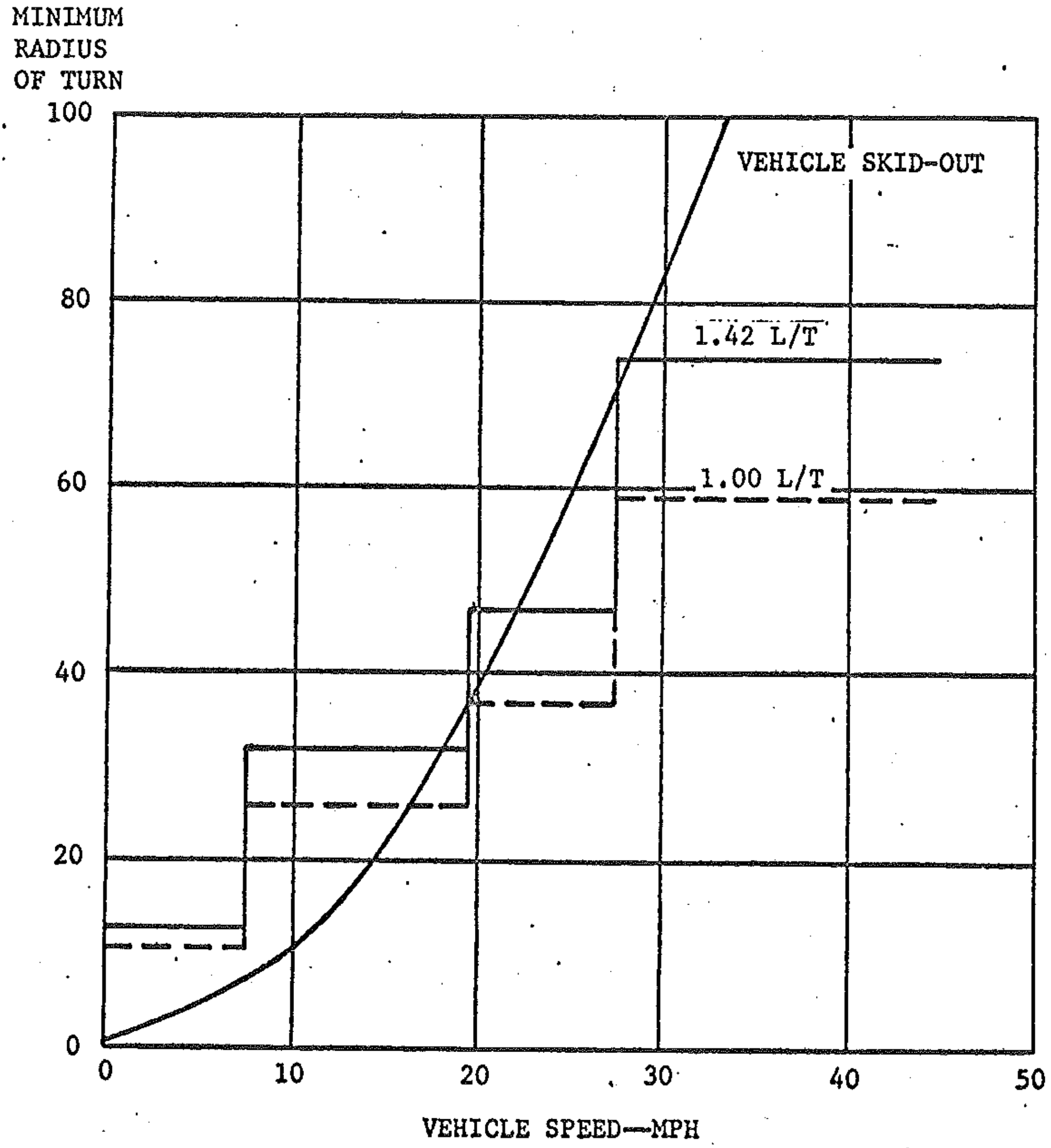


Fig. 8

determined for all maximum steer ratios.

Example

Range	Steer Ratio	$\left(\frac{V_o + V_i}{V_o - V_i}\right)$
First	3.300:1	1.87
Second	1.530:1	4.77
Third	1.346:1	6.78
Fourth	1.204:1	10.80

Now the effect of various L/T ratios on minimum radius of turn can be determined easily.

L/T Ratio	T	L	β	$\left(\frac{T + \beta L}{2}\right)$
1.42	109	154.5	.336	80.5
1.20	109	130.8	.262	69.5
1.00	109	109	.200	65.5

Range	$\left(\frac{V_o + V_i}{V_o - V_i}\right)$	$\left(\frac{T + \beta L}{2}\right)/12$			R - ft.		
		1.42	1.20	1.00	1.42	1.20	1.00
First	1.87	6.7	5.8	5.5	12.5	10.8	10.3
Second	4.77	6.7	5.8	5.5	32.0	....	26.2
Third	6.78	6.7	5.8	5.5	45.4	....	37.3
Fourth	10.80	6.7	5.8	5.5	72.4	....	59.4

These results are shown graphically, opposite, against the tracked vehicle skid-out curve.

Large L/T Ratios

From the power equations of Section IIIC it can be seen that an increase in track length, L, or a reduction in track width, T, increases the vehicle proportionality factors, β and K_1. These changes bear directly on the magnitude of the track scrubbing forces, $F_s = K_1 \mu W/2$, and the track differential, $\Delta V = (T + \beta L)\ V/12R$. The result is that the required horsepower increases significantly.

Example

Given:

$L/T = 1.42{:}1,\ 2.00{:}1$
$V_m = 20$ mph
$R = 50$ ft

L/T	T	L	β	K	K_1
1.42:1	109	154.5	.336	.608	.495
2.00:1	109	218	.553	.744	.584

Using the power equation:

$$HP_s = HP_p + HP_{sc} + (HP_{it} - HP_{fb})$$

L/T	HP_p	HP_{sc}	HP_{it}	HP_{fb}	HP_s
1.42:1	123	138	517	418	360 *
2.00:1	123	233	610	494	472

*Detail calculations shown in Section IIIC.

TEST VEHICLE CONFIGURATION

Engine		AVCR-1100-3	12V71T	12V53T
Power Rating (Net)		1300	600	550
Transmission		XHM-1500-1	X-700-1XT	X-500-1
Transmission Ratios	1st	Variable	4.16:1	5.91:1
	2nd		1.66:1	2.31:1
	3rd		1.18:1	1.64:1
	4th		.74:1	1.02:1
Final Drive Ratio		5.05	5.35	3.69
N/V Ratio (High)*		Variable	54.74	66.28
Steer System		Hydrostatic Independent Track Drive	Turbine Driven Hydrostatic Differential	Engine Driven Hydrostatic Differential
Vehicle Drive		Rear	Rear	Front

*Engine Speed to Vehicle Speed

IV. TEST PROGRAM - INVESTIGATION OF STEERING FORCES

A. Data Acquisition

1. Test Configuration

Three tracked vehicles - an M51, a T48, and an M109 - were selected for use in gathering the steer data. These vehicles, as noted opposite, were chosen because they utilized three different types of the latest in development transmissions - the XHM-1500, the X-700, and the X-500, respectively. Gear schematics of these transmissions are shown in Figs. 1-3 of Appendix A.

Vehicle loading and geometry figures, measured prior to the start of the test, are recorded in Table I in Appendix A.

2. Test Setup

The three vehicles were instrumented with strain gages, pressure transducers, speed sensors, etc, to permit electronic recording of the following data during test operation:

- Individual sprocket torque
- Individual sprocket speed
- Hydrostatic system pressure
- Range indication
- Lockup indication
- Engine speed
- Throttle position

Note: In the M51 and T48 vehicles, however, output torque was actually measured at the final drive sun gear shafts rather than the sprockets.

In addition, vehicle speed and radius-of-turn were measured and recorded manually.

3. Test Procedure

Testing of the three vehicles was conducted at General Motors Proving Ground over both level dirt and blacktop surfaces. The speeds of the vehicles were increased in 3-5 mph increments, in each transmission range, while data was recorded for the following radii of steer:

- Minimum turn
- 25 ft
- 50 ft

- 75 ft
- 100 ft
- 125 ft
- Straightaway

Note: Due to pre-emptive release of the M51 vehicle for durability testing under Contract No. DA-33-008-AMC 328(T), this vehicle could be afforded a limited amount of running on blacktop only.

4. Data Recording and Reduction

The electronic inputs from the various vehicle data sources were recorded on a seven-channel, instrumentation-type, magnetic tape recorder. A quick appraisal of the data was obtained by making an oscillograph trace record of the data. A sample trace is shown in Fig. 4 in Appendix A.

Finally the tape data (analog form) was converted to digital printout (tabular form), as shown in Fig. 5 of Appendix A, for detailed analysis purposes. In the digital printout the actual test data was broken down into twenty points per second. However, in the analysis and comparison of data, each data point consists of an average value covering a seven-second interval. This is considered to be the minimum span where test conditions were held stable.

B. Analysis and Comparison of Data

1. Radius-of-Turn and Vehicle Speed

Although attempts were made to hold radius-of-turn and vehicle speed constant during the steer operation, significant variations were found between the mechanically measured data and the data computed from output shaft speeds. Plots of indicated versus calculated speed and radius for each of the vehicles, as shown in Figs. 1 through 6 in Appendix B, show that transition points occur at approximately 9 mph and 50 feet, respectively. Above these points the indicated data consistantly lags behind the computed data.

These differences can be attributed to:

- Driver difficulty in holding a constant speed and turn with a tracked vehicle

- Vehicle drift

2. Track Forces

In the lower ranges of speed and radius of turn, both the inner and outer track forces decrease as speed and radius increase (as shown in Figs. 7 through 11 of Appendix B). However, in the higher ranges, a "leveling off" of track force occurs. This characteristic substantiates present theories that steering of tracklaying vehicles should be a speed-controlled function rather than a force-controlled function. This is particularly true if the trend to higher speeds (above 40 mph) is to continue. At such speeds, where force requirements remain essentially unchanged, only a differential speed sensing system can provide precise steer control.

3. Vehicle Slewing Forces

Figs. 12 and 13 in Appendix B show slewing force versus radius-of-turn for the T48 and M109 vehicles (the two for which the most data was collected). A comparison of these figures shows that the magnitudes of the slewing forces for each vehicle are approximately equal with those of the lighter M109 being slightly lower.

Slewing force, being on a pounds per ton basis, should be equal from one type and size of vehicle to another if all vehicle characteristics are equal. On the basis of the L/T ratios for the two vehicles, however (1.42 for the M109 and 1.36 for the T48), the M109 slewing forces should be the higher of the two. The discrepancy here must be attributed to other differences in the vehicles' characteristics (ref: Table I in Appendix A).

4. Transmission Hydrostatic Pressures

The translation of hydrostatic steer pressures to transmission torque and then on to track force is another tool for calculating vehicle steer functions. To verify this statement, the slewing forces in the M109 vehicle were determined from recorded hydrostatic pressure, recorded sprocket torque, and Merritt's formula. A comparison of data for third and fourth range operation at a 50-radius is shown below.

Speed-MPH	Slewing Force - Lb/Ton by Hydrostatic Pressure	Slewing Force - Lb/Ton by Sprocket Torque	Slewing Force - Lb/Ton by Merritt's Formula
Third Range			
5	579	630	581
10	562	607	581
15	525	558	581
20	437	429	581
Fourth Range			
5	575	639	581
10	549	609	581
15	529	575	581
20	422	418	581

The data indicates a maximum disagreement of 10 percent between the pressure and torque generated figures. The Merritt-formula figures, however, disagree by as much as 28 percent at the relatively low speed of 20 mph. This is due to the fact that Merritt's formula, $F = K\mu W$, does not make provision for the reduction of force due to reduction in speed or increase in radius-of-turn.

The apparent advantages to be gained from the resulting agreement between hydrostatic pressure and sprocket torque include:

- Recorded track force data can be used to accurately estimate hydrostatic size.
- Hydrostatic pressure test data can be used to determine, theoretically, the steer capability of a proposed installation.
- Recording hydrostatic pressure is a less complex task than recording sprocket torque for the purpose of evaluating vehicle steer.

5. Track Length-over-Width Ratio (L/T)

Wide but short vehicles with their small L/T ratios are easier to turn than long and narrow vehicles having large L/T ratios. Conversely though, vehicles with small L/T ratios suffer a loss of steering stability.

An analysis of the steer data was conducted in an attempt to confirm the effect of length over width and thus verify the ratio band limits where vehicles would steer and track with the greatest degree of success. However, the wide scatter of points, apparently due to a lack of sensitivity of the instrumentation, prohibited this. The problem was also compounded by variations in:

- Type and size of track
- Track geometry
- Number of bogies
- Bogie loading patterns
- Length of track between bogies

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XHM-1500 TRANSMISSION SCHEMATIC

(REF: ASSEMBLY DRAWING DAX 8515)

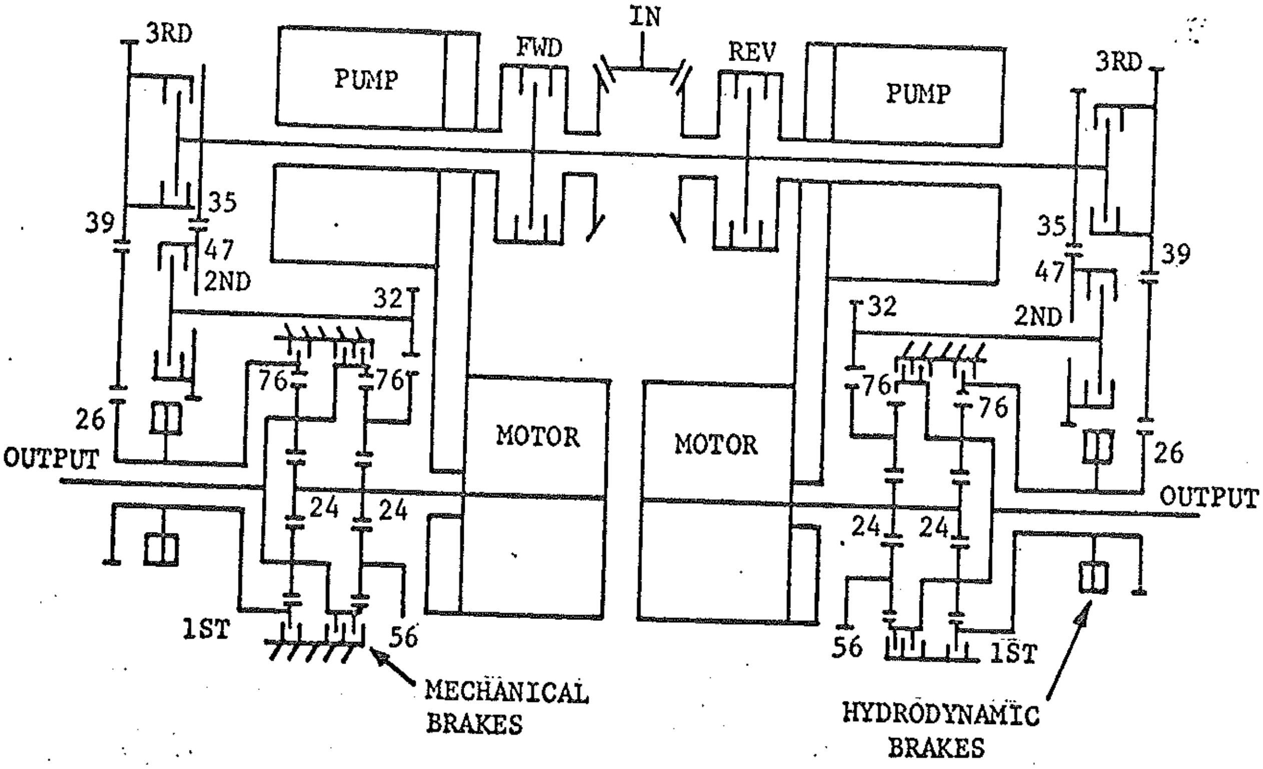


Fig. 1

X-700 TRANSMISSION SCHEMATIC

(REF: ASSEMBLY DRAWING DAX-5450)

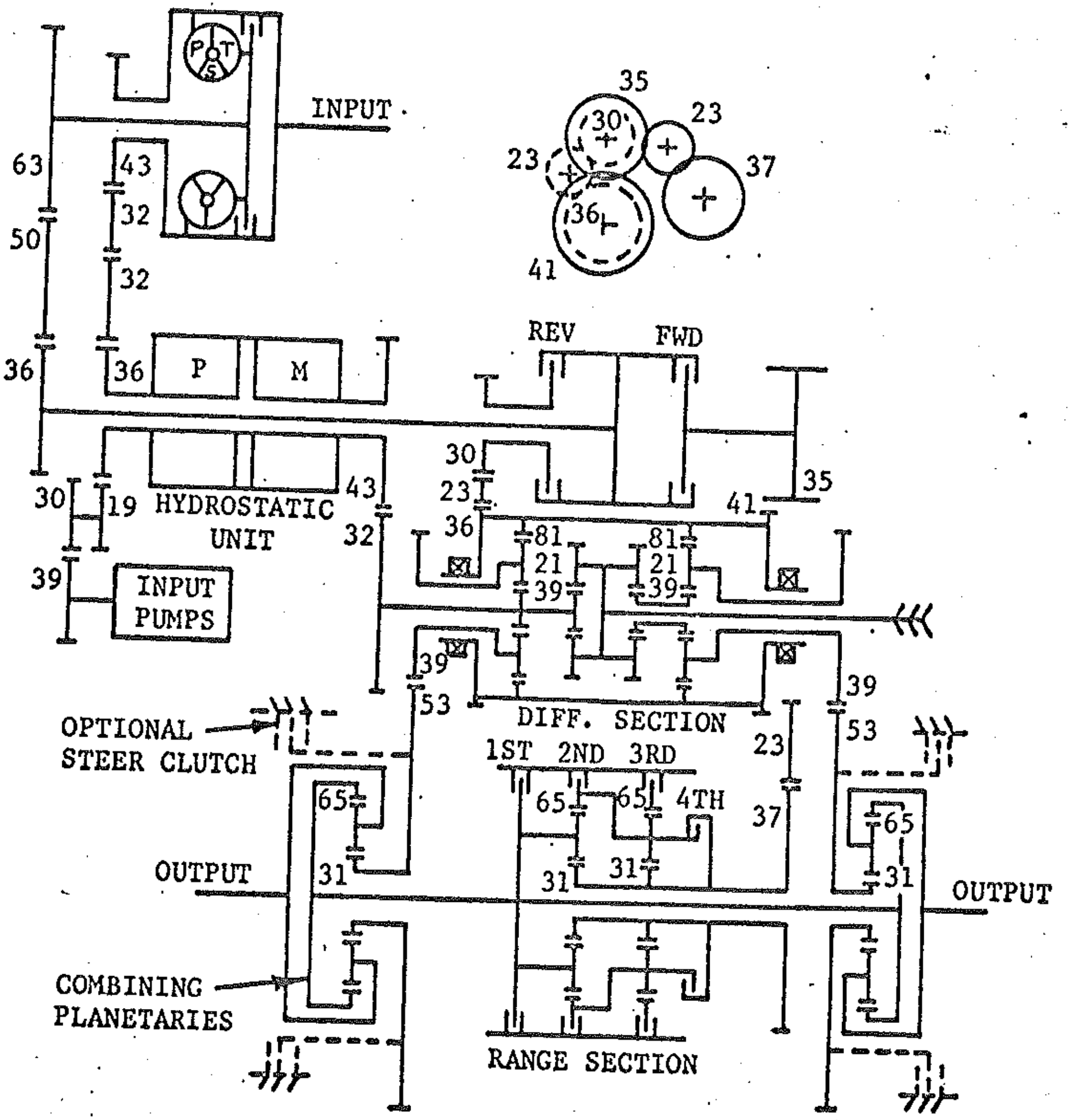


Fig. 2

X-500 TRANSMISSION SCHEMATIC

(REF: ASSEMBLY DRAWING DAX-8400)

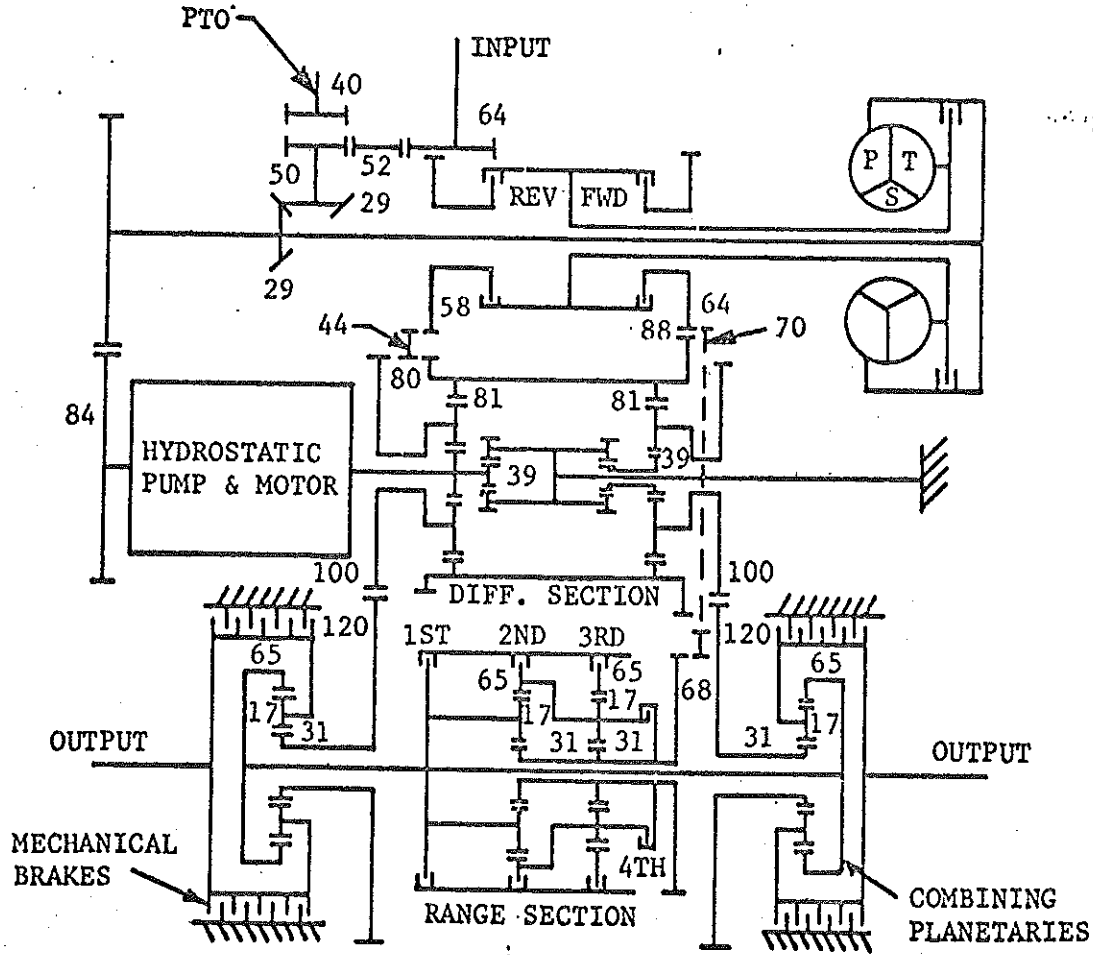


Fig. 3

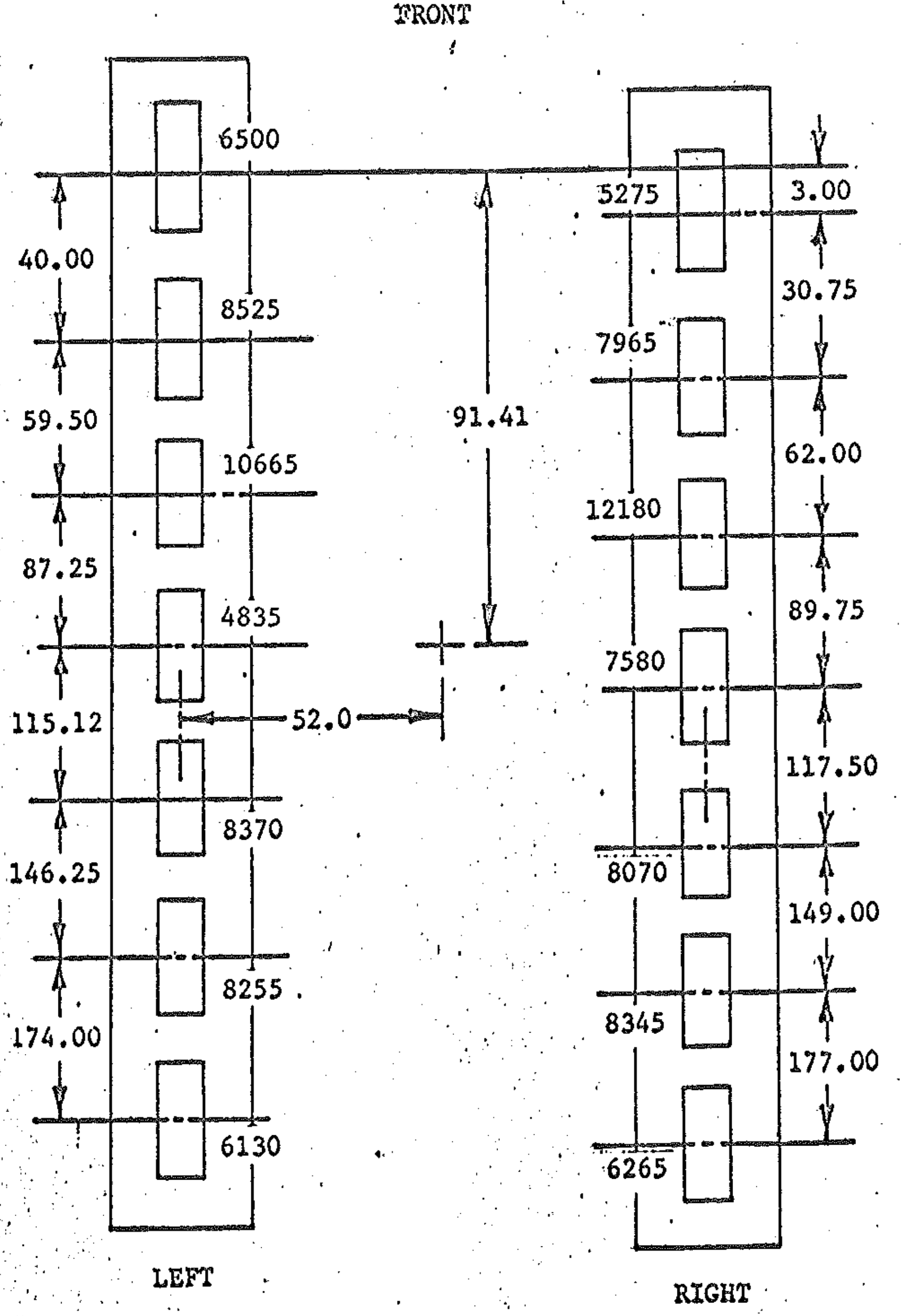
		M51	T48	M109
Gross Vehicle Weight, Lb.	116000	106460	88480	52180
Vehicle Center of Gravity, In.		p47	p48	p49
No. of Bogies, Each Side		7	6	7
Location & Loading of Bogies		p47	p48	p49
Track Length on Ground, In.	150	174	158	154.5
Track Center-to-Center Distance, In.	108	116	116	109
Track Width, In.	16.8	28	28	15
Sprocket Pitch, In.		7.125	6.094	6.0
No. of Sprocket Teeth		11	11	10

Table I

8

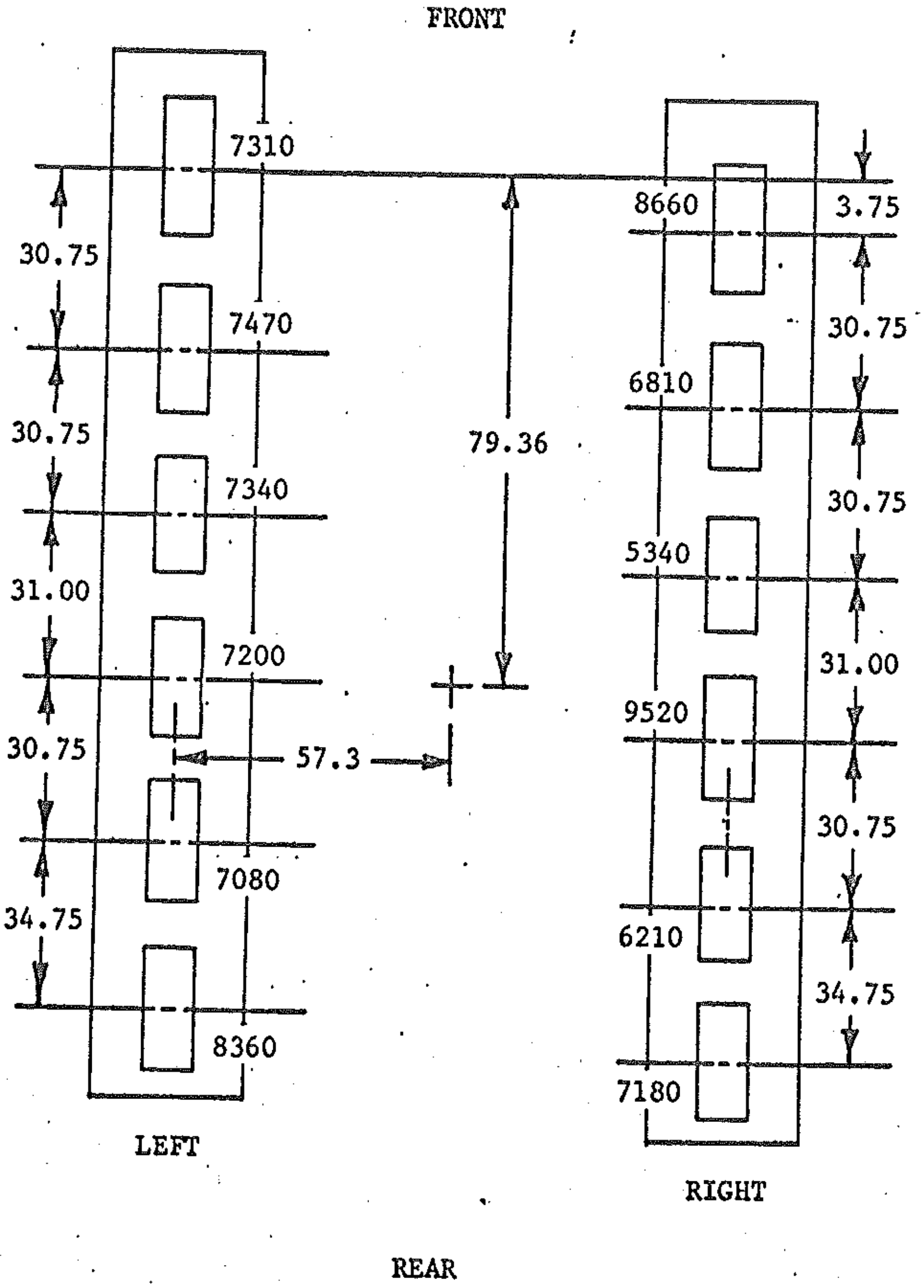
VEHICLE MEASUREMENTS

M51



VEHICLE MEASUREMENTS

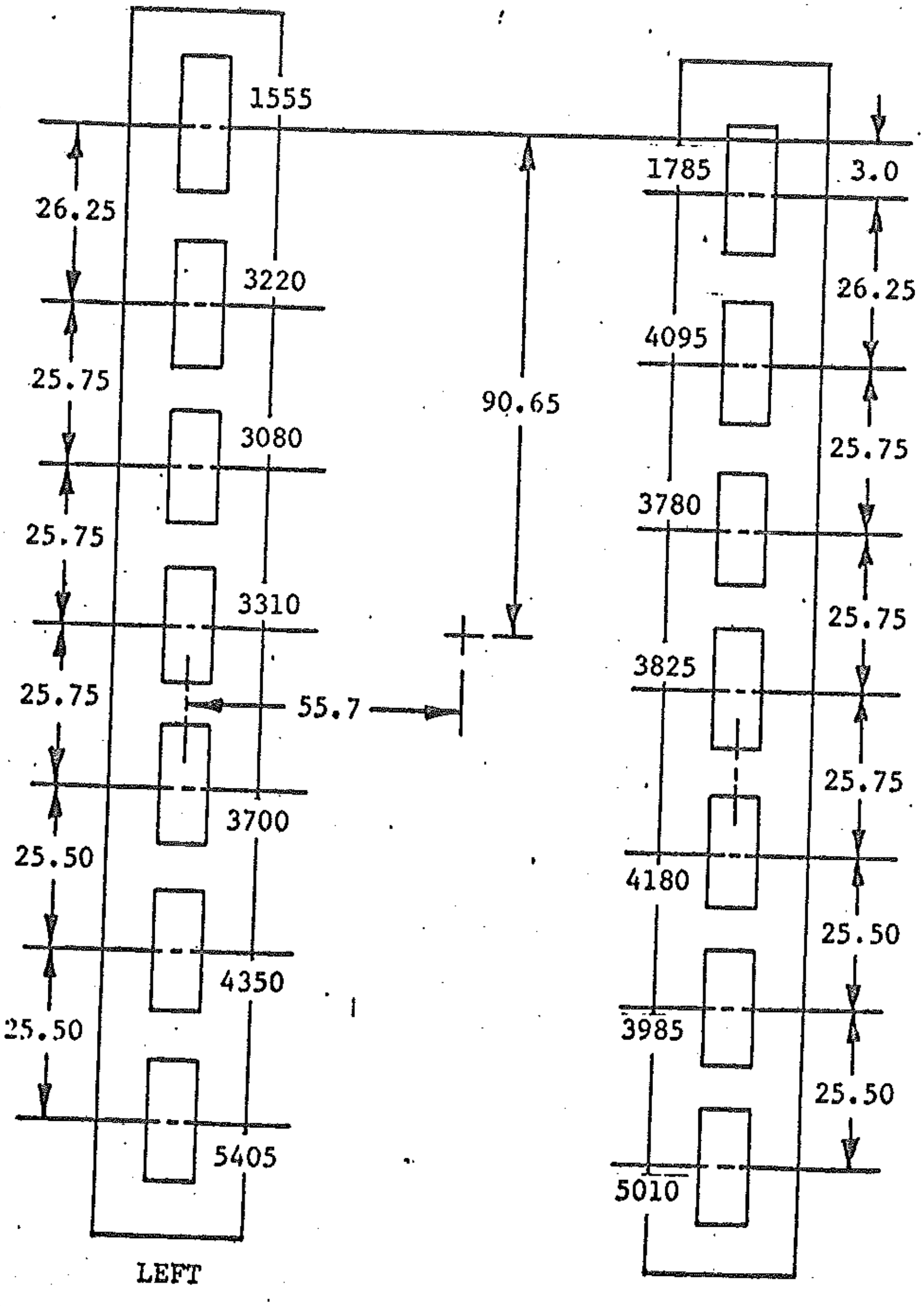
M48



VEHICLE MEASUREMENTS

M109

FRONT



REAR

DATA TRACE

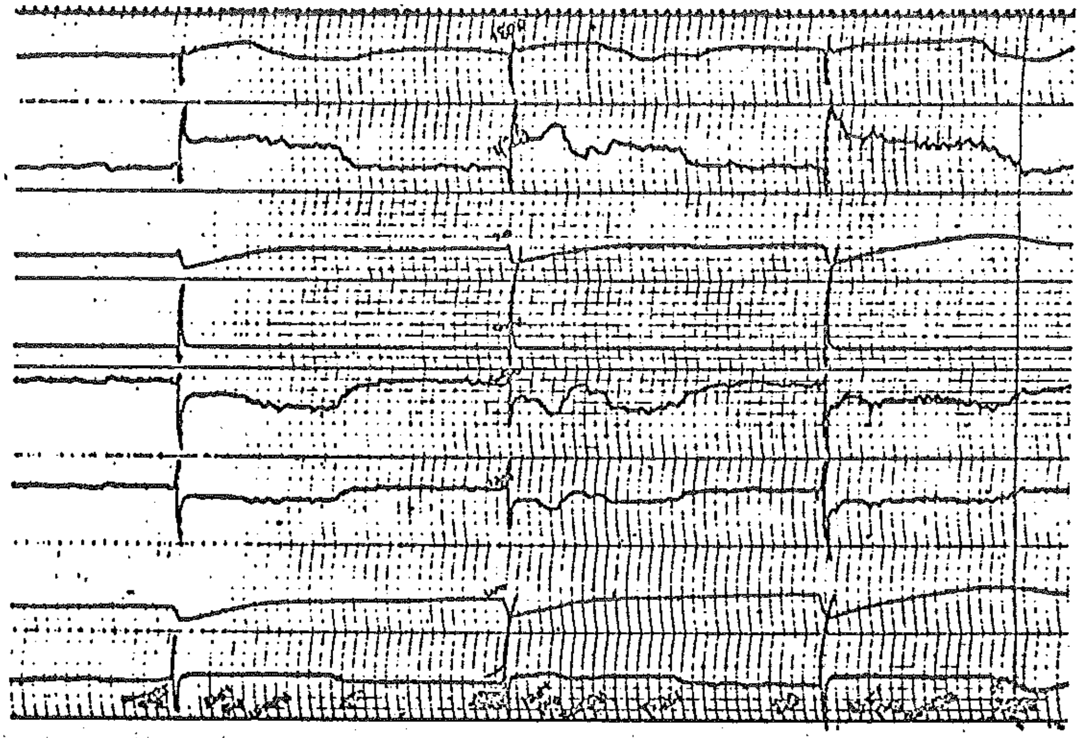


Fig. 4

DIGITAL PRINTOUT

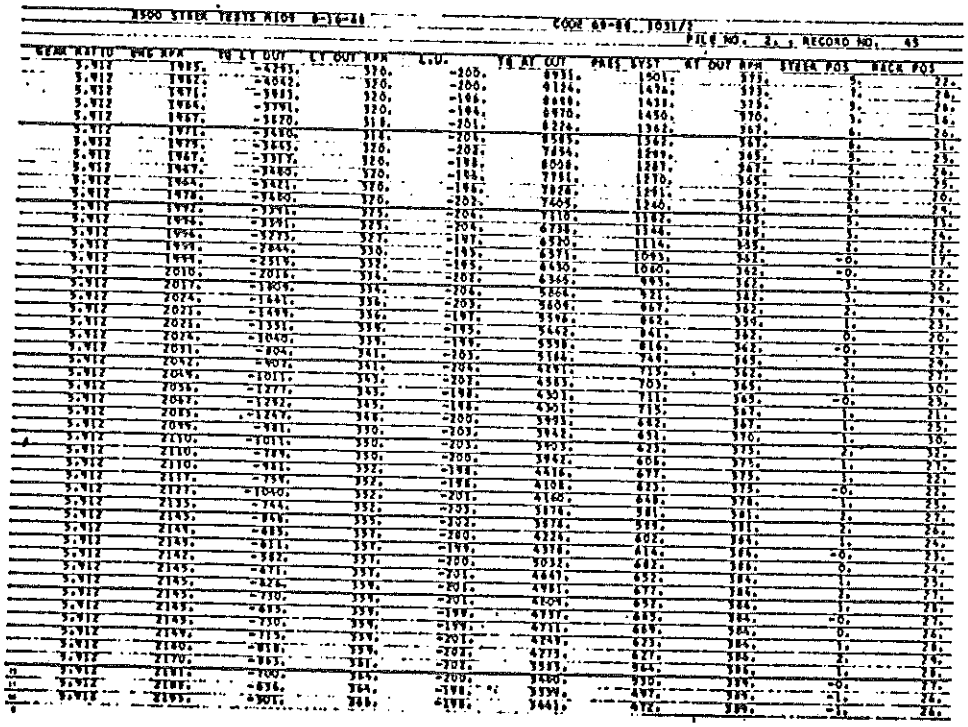


Fig. 5

APPENDIX B

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SPEED DEVIATION

M51 VEHICLE, ASPHALT SURFACE

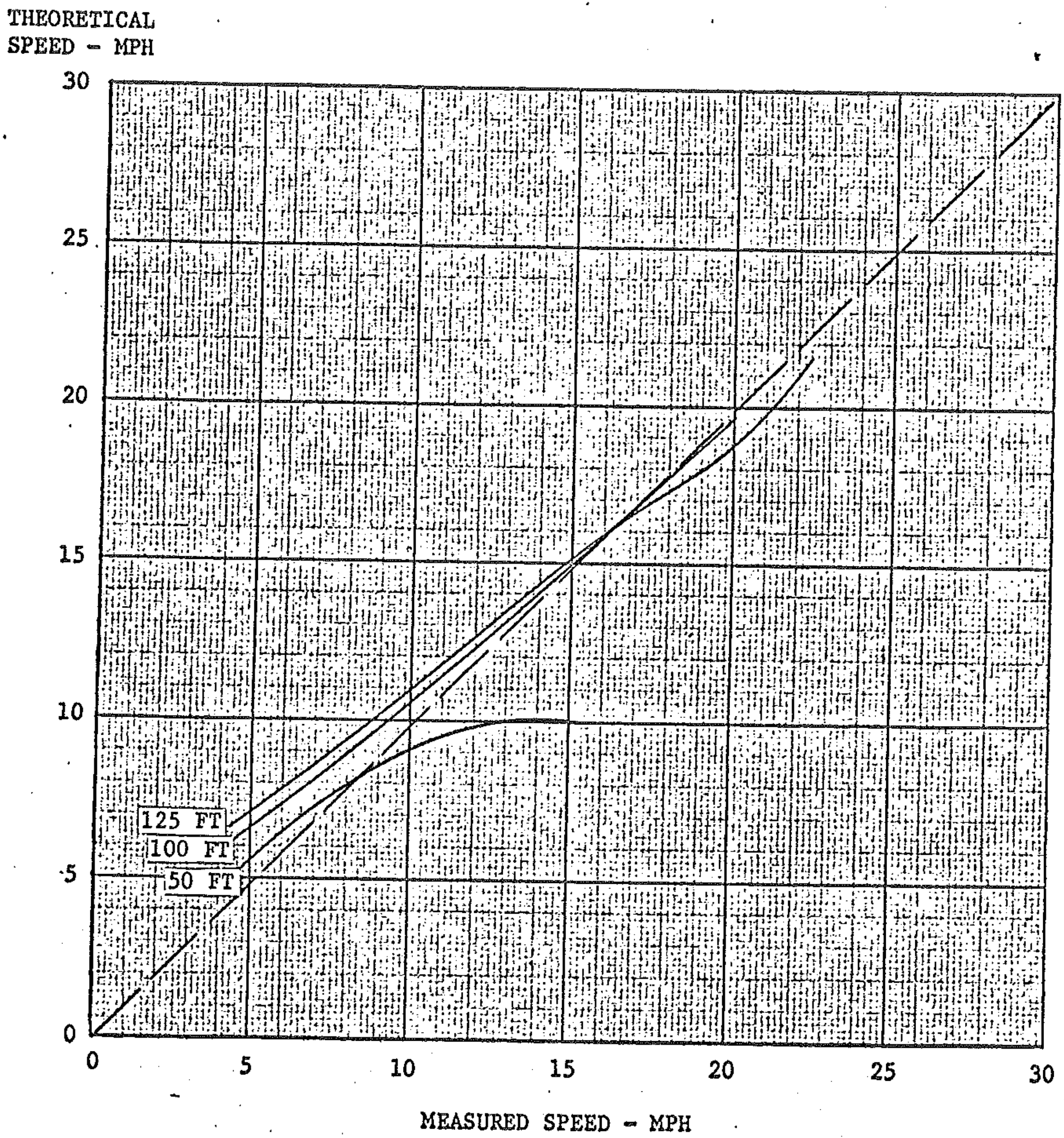

Fig. 1

SPEED DEVIATION

T48 VEHICLE, ASPHALT SURFACE

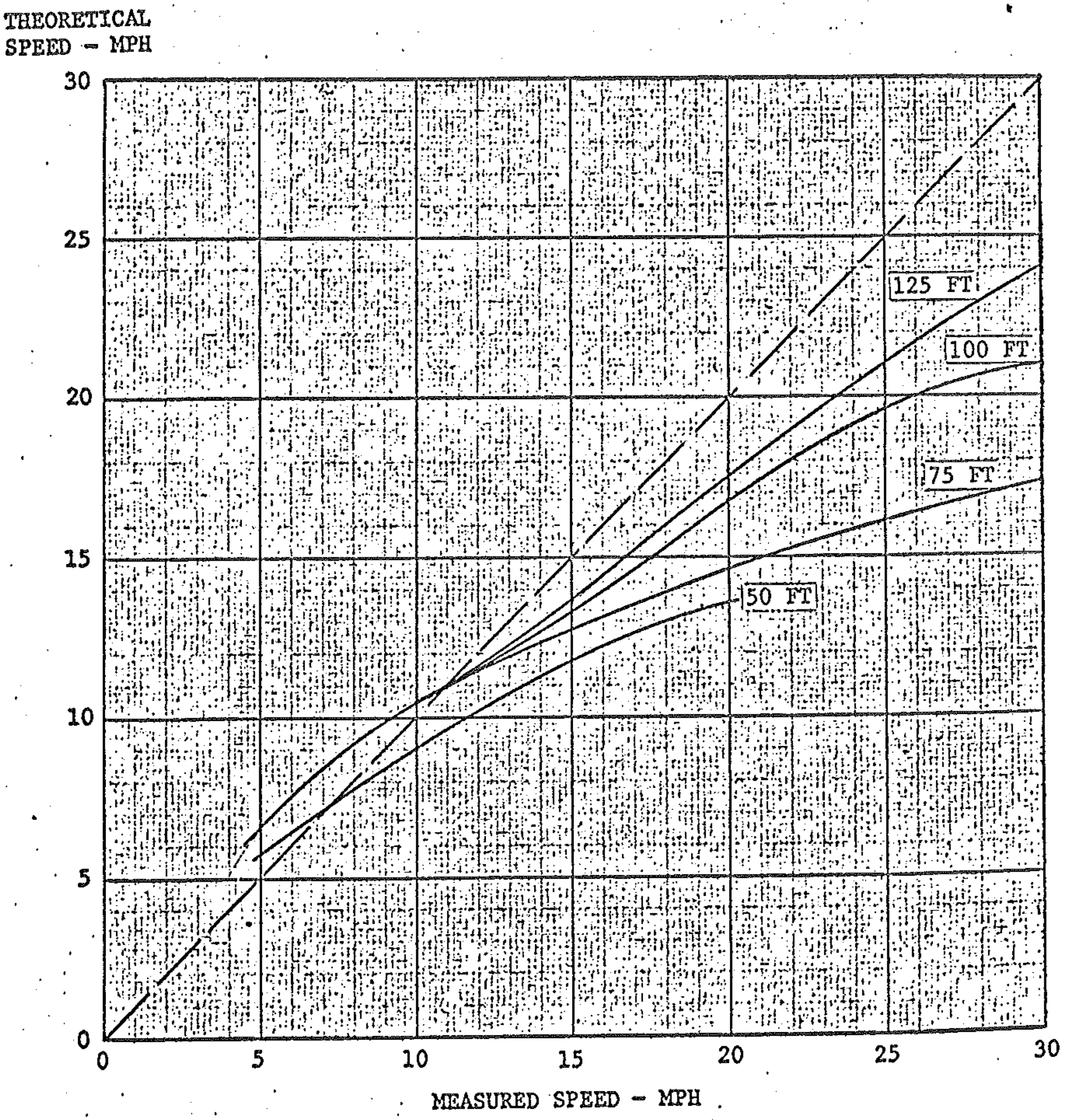

Fig. 2

SPEED DEVIATION

M109 VEHICLE, ASPHALT SURFACE

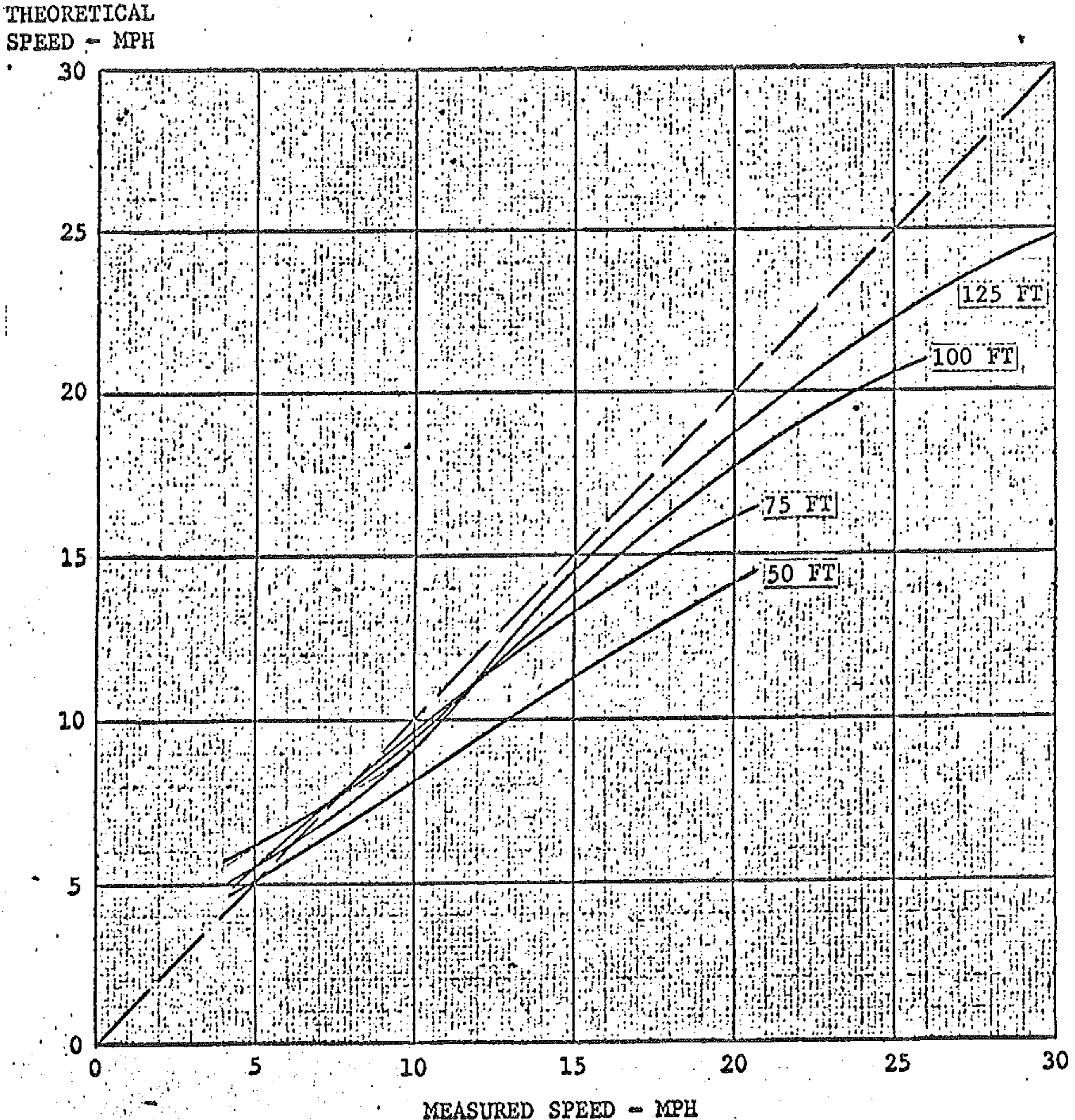

Fig. 3

RADIUS-OF-TURN DEVIATION

M51 VEHICLE, ASPHALT SURFACE

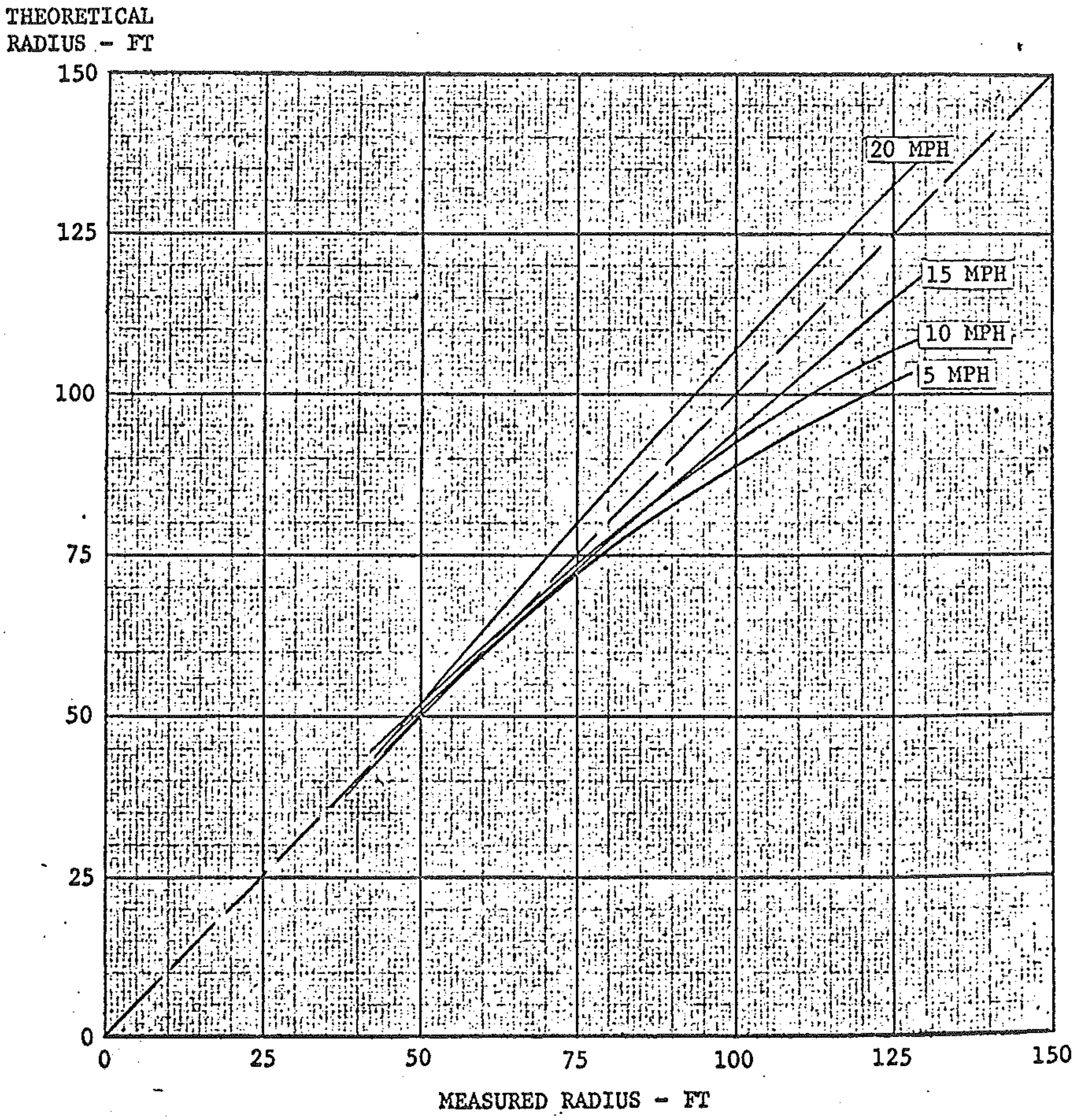

Fig. 4

RADIUS-OF-TURN DEVIATION

T48 VEHICLE, ASPHALT SURFACE

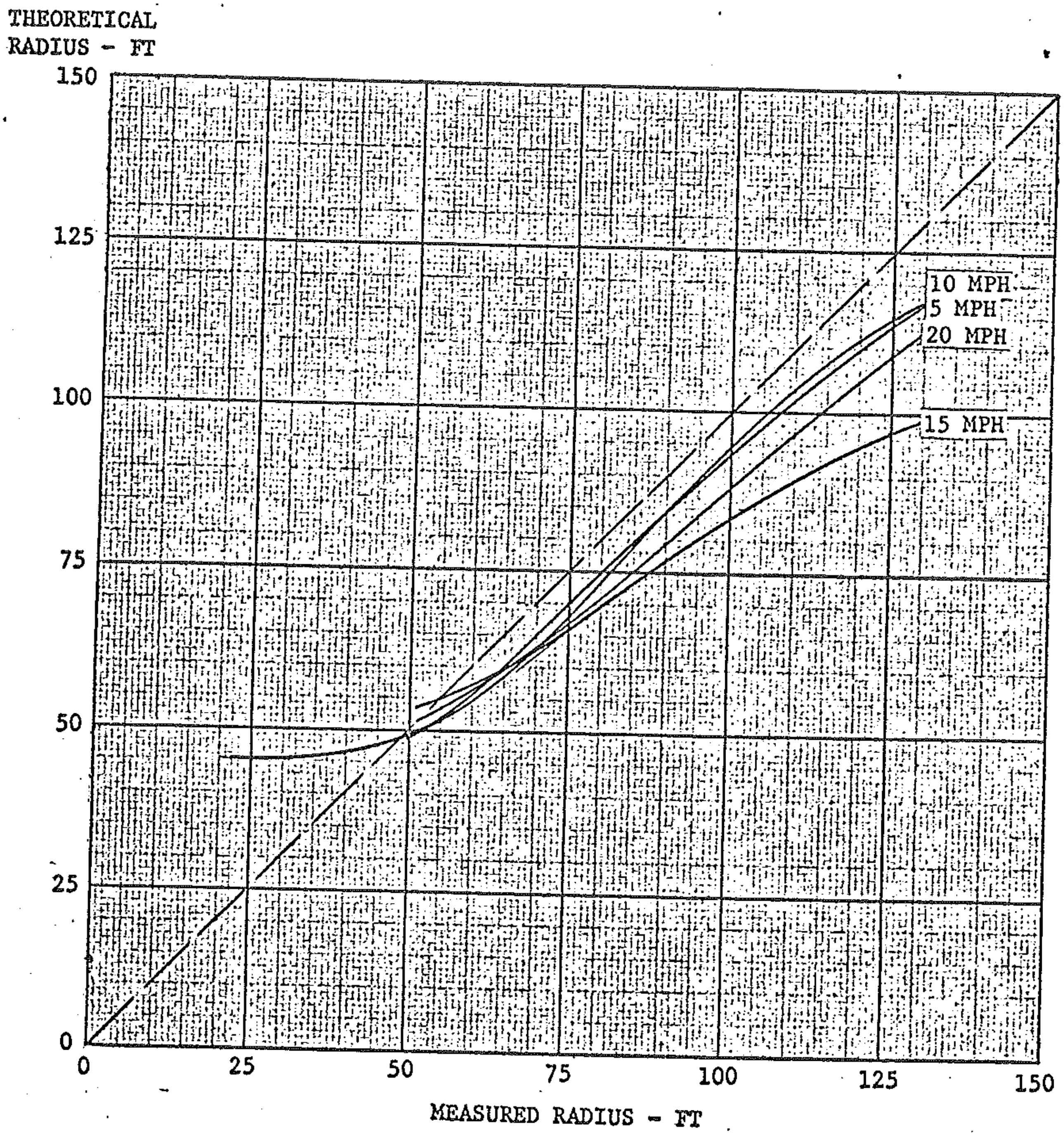

Fig. 5

RADIUS-OF-TURN DEVIATION

M109 VEHICLE, ASPHALT SURFACE

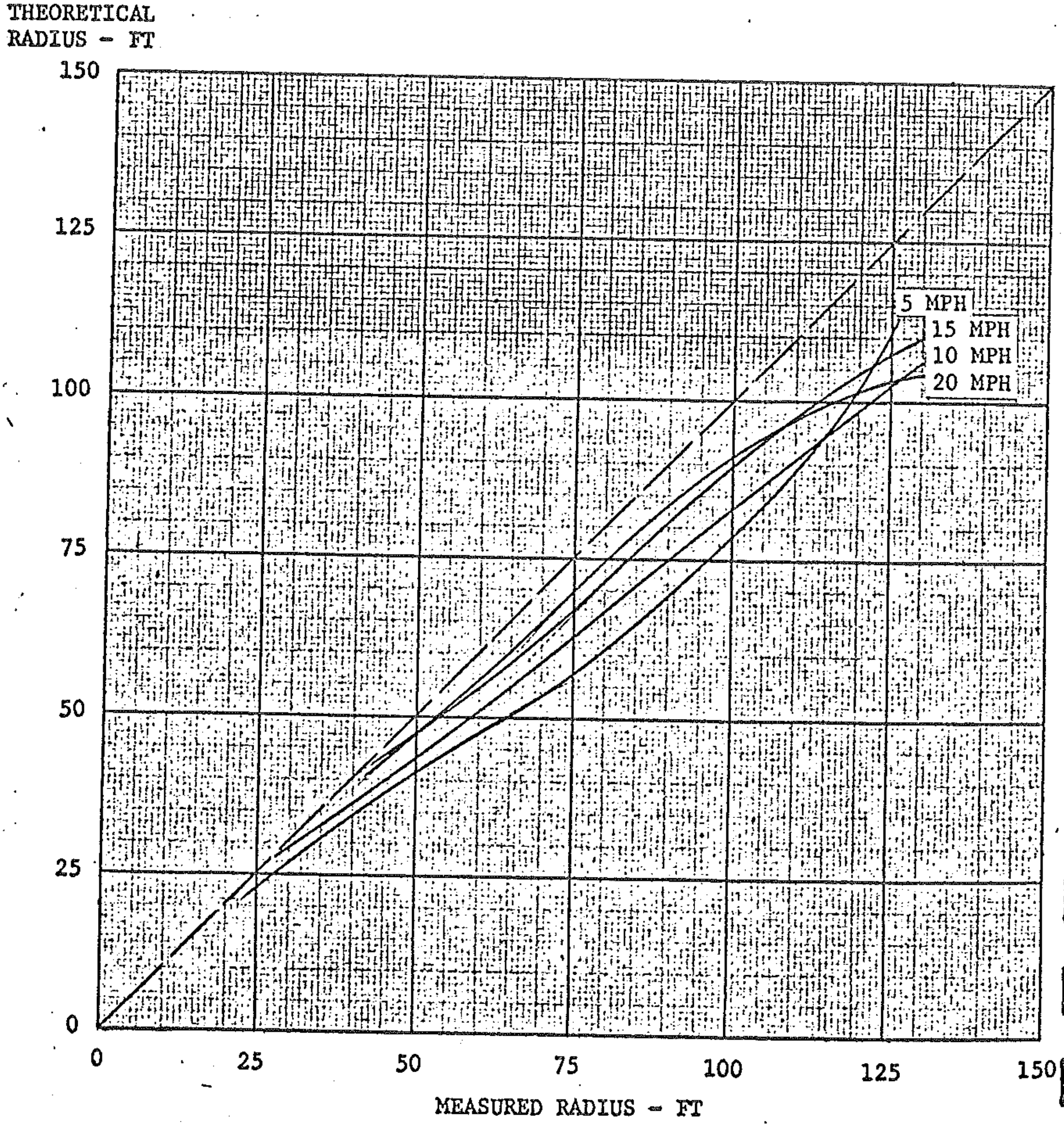

Fig. 6

46000 –

4600-5220 = 40780

$\frac{40780 \times 4.82}{575}$ = 6157P

TRACK FORCES

M51 VEHICLE, ASPHALT SURFACE

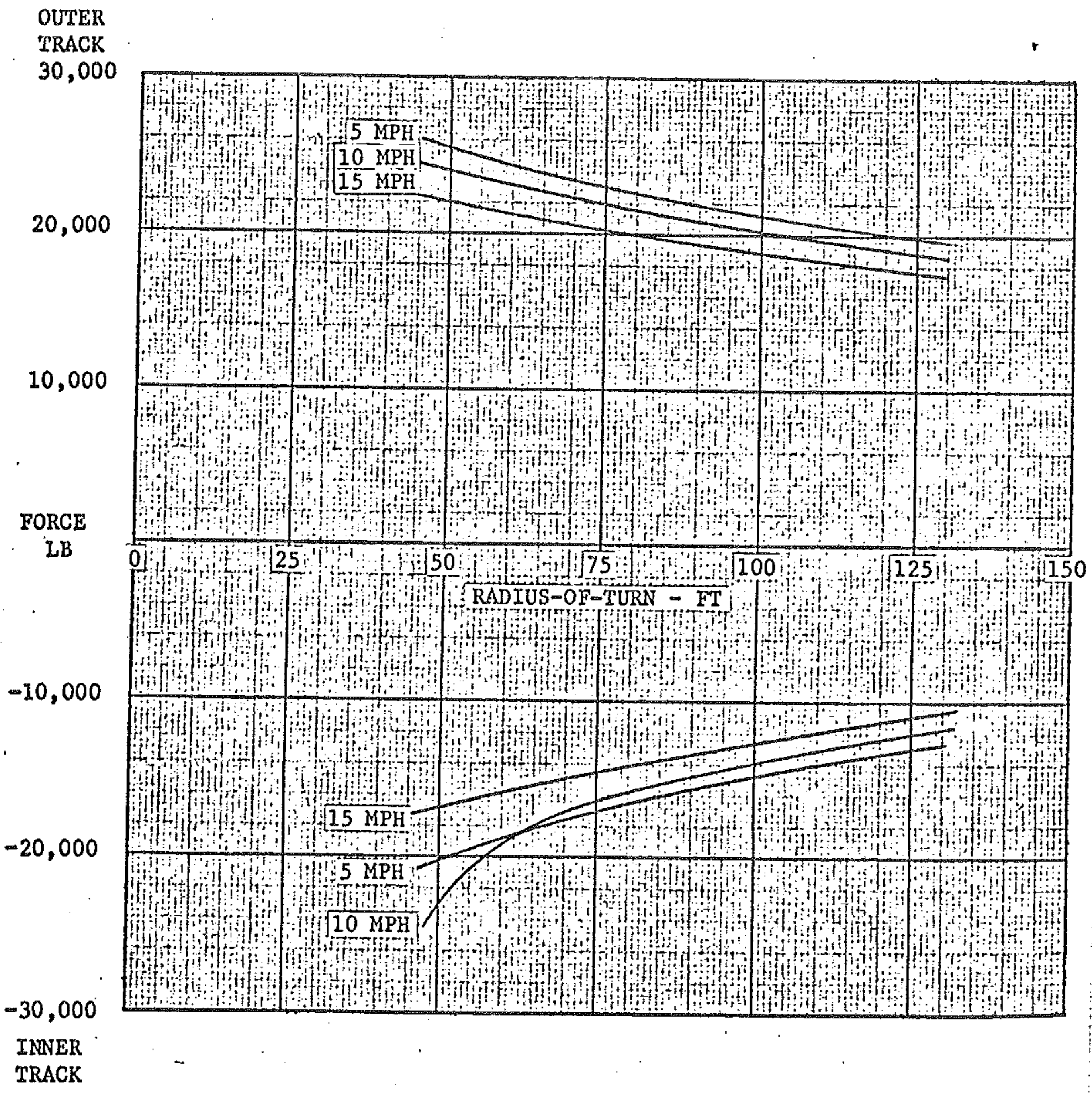

Fig. 7

TRACK FORCES

T48 VEHICLE, ASPHALT SURFACE

5400

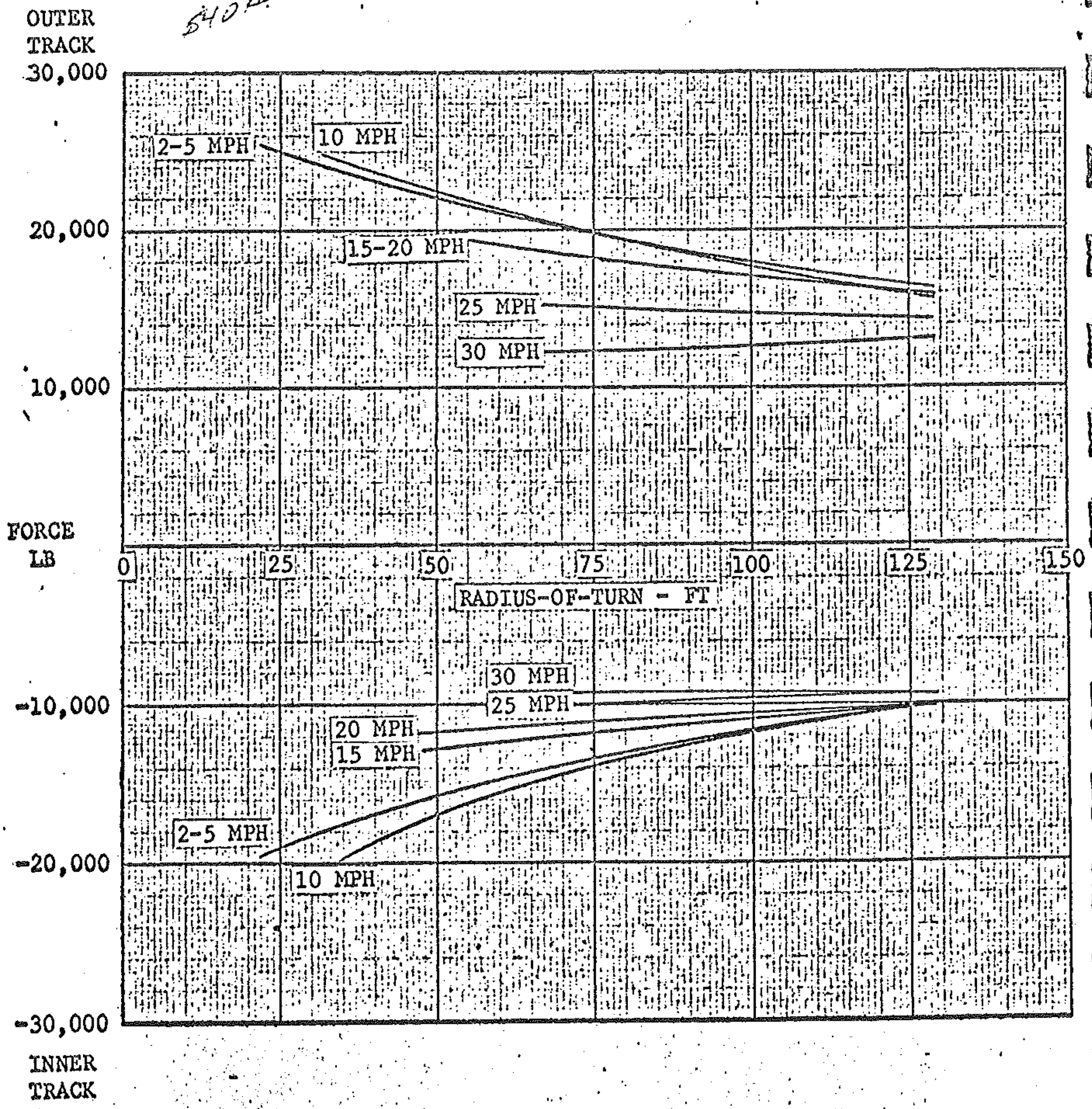

Fig. 8

TRACK FORCES

T48 VEHICLE, DIRT SURFACE

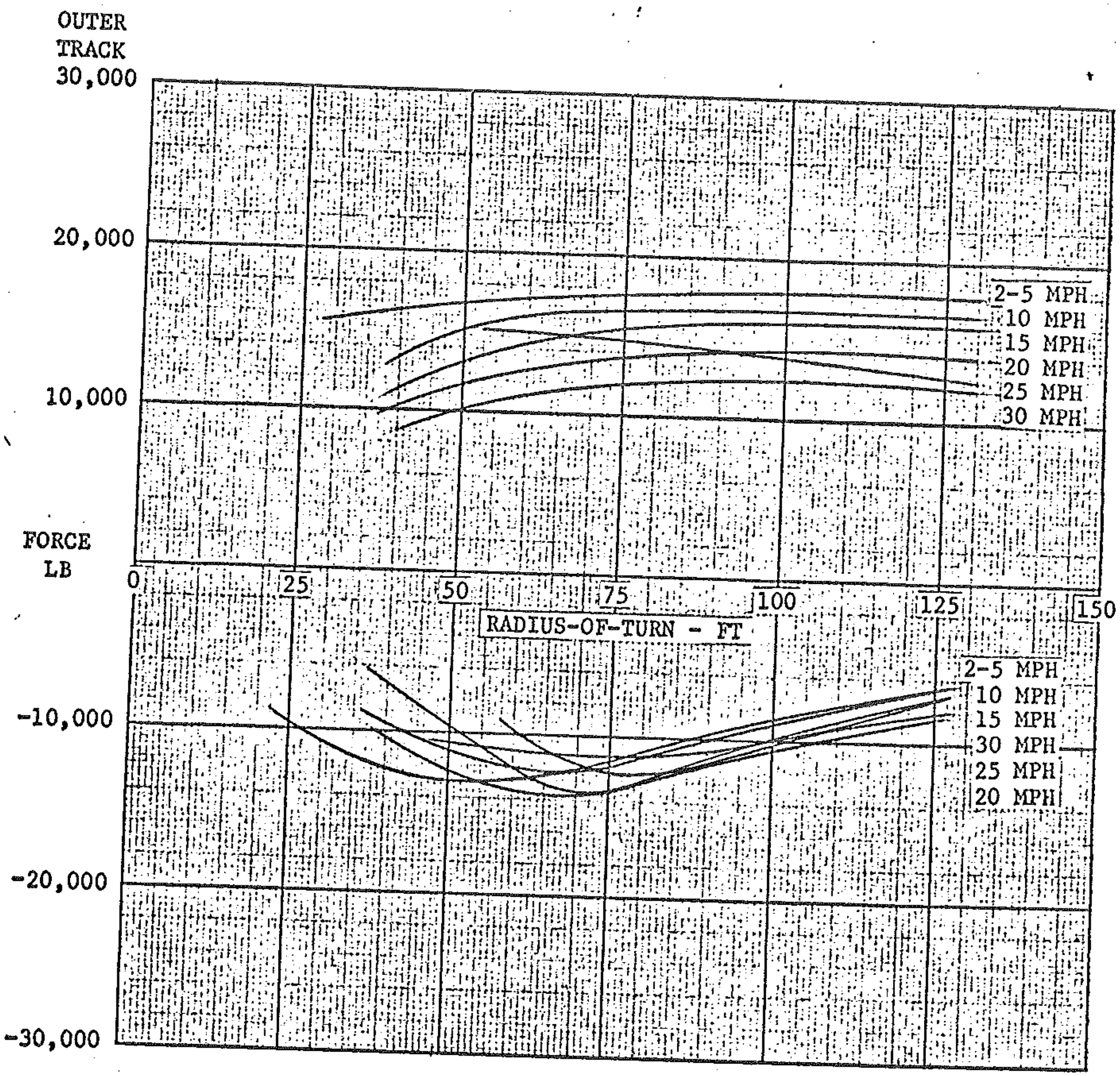

Fig. 9

TRACK FORCES

M109 VEHICLE, ASPHALT SURFACE

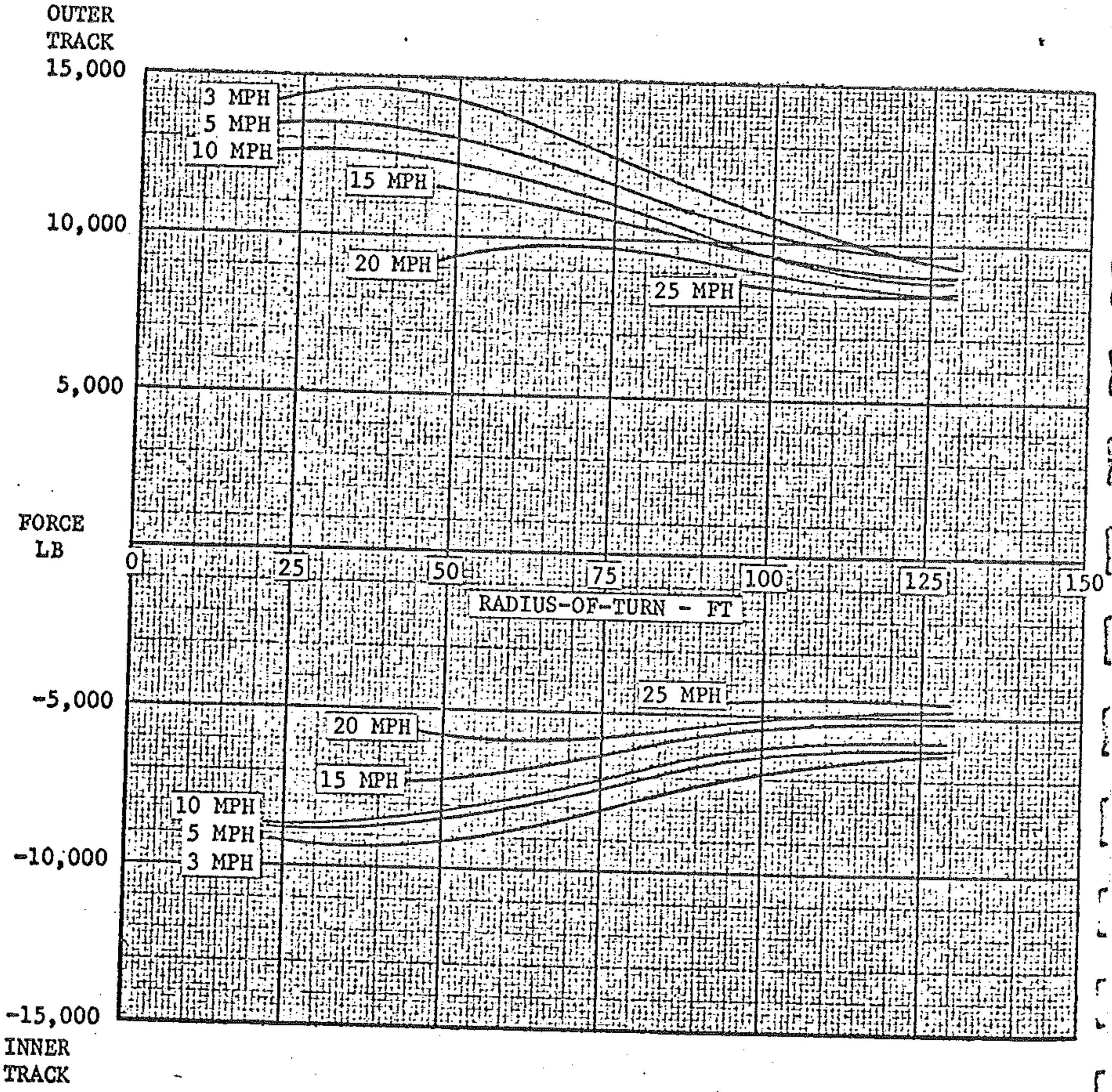

Fig. 10

TRACK FORCES

M109 VEHICLE, DIRT SURFACE

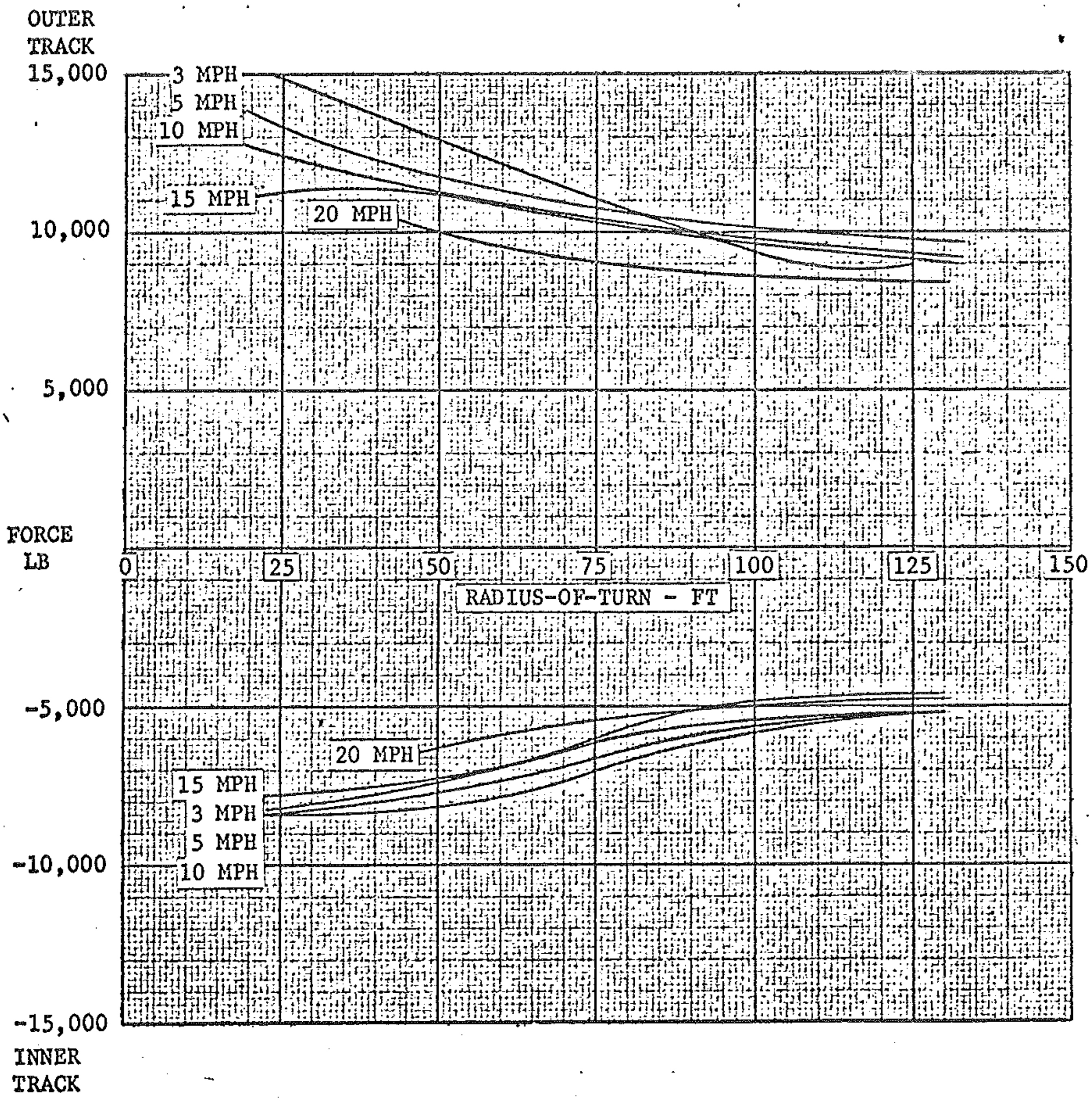

Fig. 11

SLEWING FORCES

T48 VEHICLE, ASPHALT SURFACE

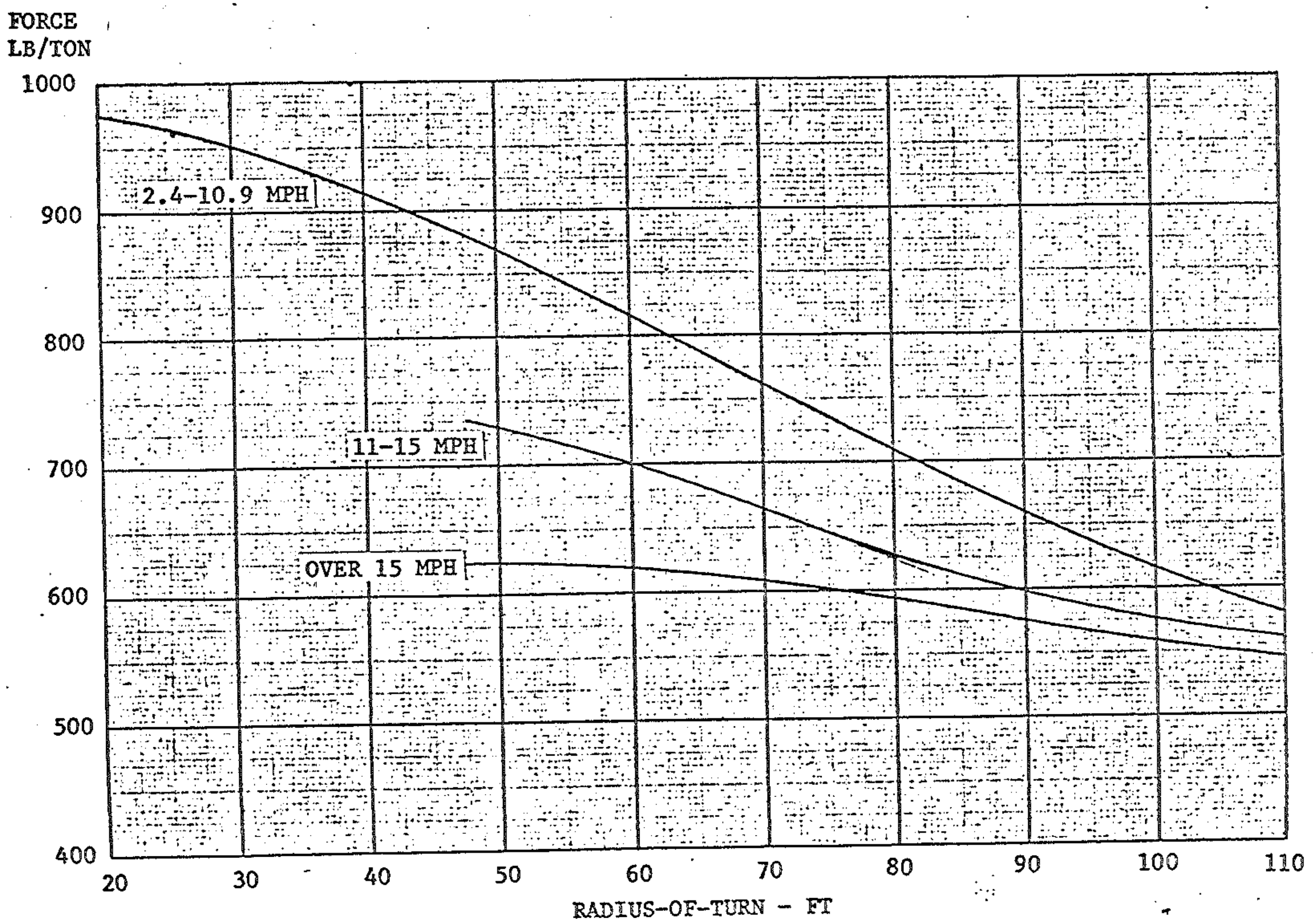

Fig. 12

SLEWING FORCES

M109 VEHICLE, ASPHALT SURFACE

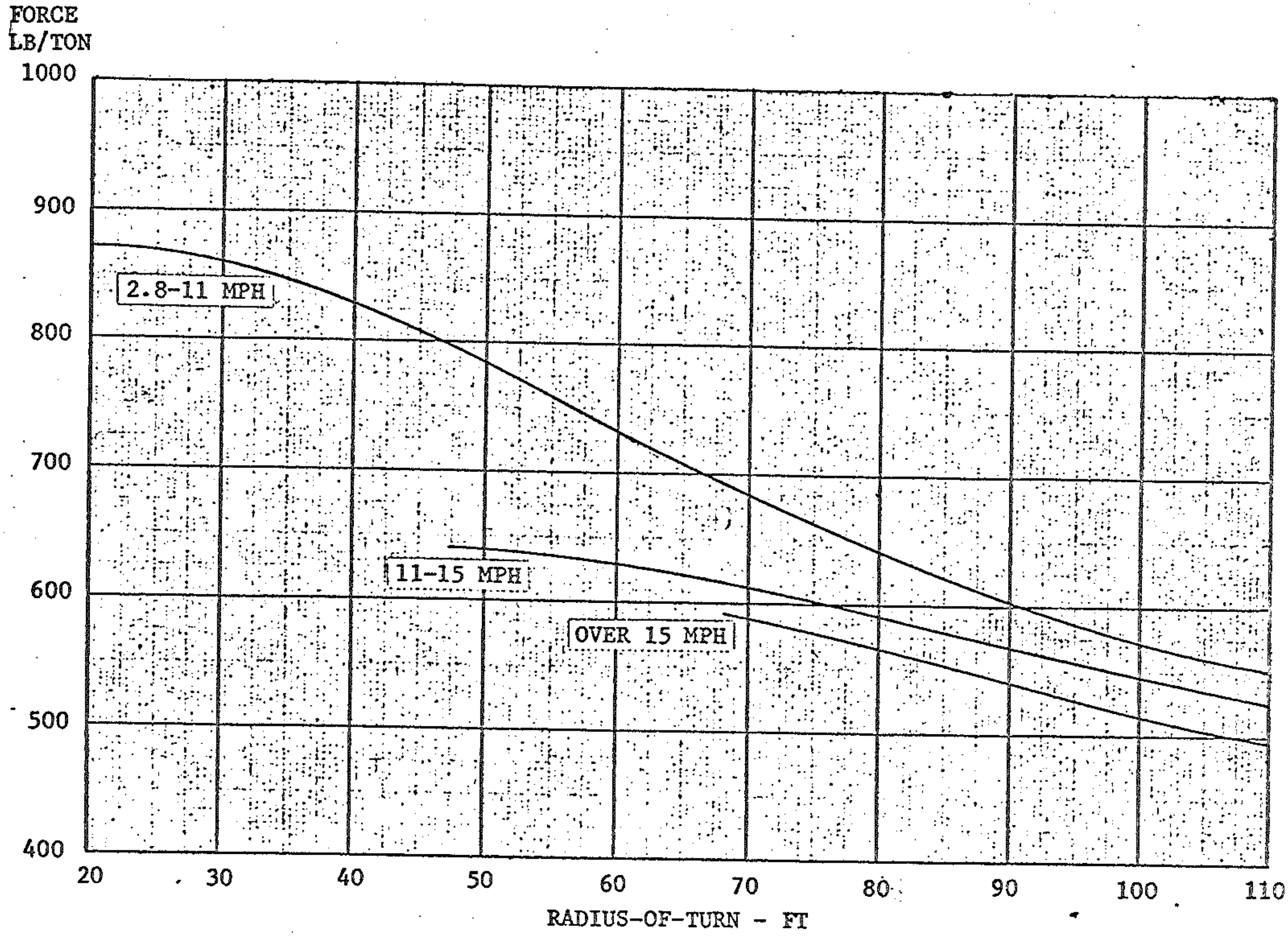

Fig. 13

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13. ABSTRACT

Since concepts involved in designing tracklaying vehicle steer systems and calculating steer characteristics are complex, efforts are continuously made to review recent advancements and reevaluate existing knowledge.

In a program to review the current state-of-the-art in steer system design, three current transmissions concepts were compared by means of vehicle performance tests. The data gathered was reduced electronically, and the results were analyzed for new findings. The results of the program confirmed present theories that, at higher vehicle speeds (above 40 mph), a "leveling off" of track force requirements occurs. This would indicate that steering of tracklaying vehicles should be a speed-controlled function rather than a force-controlled function.

Also as part of the program, the methods for calculating track forces, speeds, power requirements, and radius-of-turn were reviewed, and various constants and variables were confirmed or revised as test data indicated. That phase of the program resulted in the preparation of this manual to aid vehicle design engineers in calculating expected steer performance.

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